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# NOTES ON

# Railway Mechanical Engineering

by

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# NOTE

The material in the following pages is the outgrowth of lecture notes prepared for the use of fourth—year students in Nanyang University. It is assumed that the student in fully prepared in mathmatics, thermodynamics, mechanics, (theoretical and applied), and machine and engine design, as covered by in struction given in this school. Illustrative problems covering all phases of the subject will be used in conjunction with the printed notes. The author will be grateful for having his attention called to such errors and inconsistencies are may be found.

SHANGHAI, SEPT. 1, 1923.

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# Notes on Railway Mechanical Engineering.

## GENERAL

I. SCOPE. The subject embraces the study of rolling stock as to its design, construction and operation, also the arrangement and equipment of repair shops and terminals for railway stock. Many other closely related matters are considered in other branches of engineering and will be only briefly mentioned. The steam locomotive, as the characteristic mechanical feature of the railway will be given first attention.

II. HISTORICAL. The first real locomotive in the modern sense, capable of pulling cars was built in 1803 by Trevethick, a Cornish mineforeman. The first to incorporate the usually accepted modern principles was built by Stephenson in 1829 for the Liverpool and Manchester Railway—the "Rocket." This engine used coal or coke fuel in a rectangular firebox with grate bottom, a multitubular boiler (firetube), the draft being furnished by exhausting the steam into the smoke stack. The engine was provided with two inclined cylinders, and steam distribution was controlled by valve-gears with provision for variable cutoff and reversing. Important operating principles since adopted are compounding (1875——little used at present), superheating (1900), articulating (1880——little used) and feedwater heating which is now coming into practice. Attempts have been made to use the water tube boiler, the condenser, and the turbine blower, but none have been fully successful. The Diesel locomotive and the turbine locomotive are now being tried, but are far from perfected.

III. THE LOCOMOTIVE POWER PLANT. In its simplest form the locomotive represents the simplest possible power plant (steam only considered), merely a furnace, boiler, and two simple reciprocating engines. In its more highly developed state (the articulated engine being the most advanced) the following auxiliaries may be present in addition to the use of compound cylinders: superheater, feedwater heater, reaheater. The condenser is not practicable. The following list covers the most important parts of this power plant:

- Boiler
- Firebox or furnace
- Grate
- Ashpan
- Superheater flue
- Tube
- Smoke box or smoke—arch
- Smoke stack or chimney
- Steam dome
- Cylinder
- Piston
- Piston rod
- Crosshead
- Slide bars or guides
- Frame
- Main or Connecting Rod
- Side or parallel rod
- Driving wheel
- Throttle or regulator
- Dry pipe or delivery pipe
- Superheater header
- Front end steam pipes
- Exhaust nozzle
- Pilot
- Bumper or buffer beam
- Coupler
- Draw bar
- Cab (“in the cab” = “on the footplate”)

Study the variation in construction of these parts, and the parallel parts is the stationary power plant. What is the parallel part to the flywheel of the stationary engine? of the locomotive side rod?

The great disparity in size between the steam locomotive and a stationary plant capable of developing the same power is immediately apparent. How much space is required to house a stationary power plant of 100 H. P. capacity? How much to house a locomotive of 2,000 ft. heating surface? (capacity about 1000 H. P.) The locomotive boiler is driven from three to ten times as hard as the stationary boiler. What controls the rate of combustion in any boiler? What was the draft you found in your power-plant tests? Fifteen inches is not an extreme value for a locomotive boiler.

IV. CLASSIFICATION. Locomotives are usually classified according to their most prominent external feature, namely the number and arrangement of the wheels. Letting “o” represent a carrying wheel or truck wheel and “O” a driving wheel the following classification should be noted. (The front of the engine is at the left in each case.)

| Wheel Arrangement | Numerical Classification | Name | Service |
|---|---|---|---|
| 00 | 040 | Four-wheel switcher | Shunting |
| 000 | 060 | Six-wheel switcher | Sunting |
| 0000 | 080 | Eight-wheel switcher | Shunting |
| 00000 | 0100 | Ten-wheel switcher | Shunting |
| o000 | 260 | Mogul | Mixed and Goods |
| o0000 | 280 | Consolidation | Goods |
| o00000 | 2100 | Decapod | Goods |
| o000o | 262 | Prairie | Goods and Mixed |
| o0000o | 282 | Mikado | Goods |
| o00000o | 2102 | Santa Fe | Goods |
| oo00 | 440 | Eight Wheel | Passenger |
| oo000 | 460 | Ten-wheel | Passenger |
| oo0000 | 480 | Twelve-wheel | Goods |
| oo00o | 442 | Atlantic | Passenger |
| oo000o | 462 | Pacific | Passenger |
| oo0000o | 482 | Mountain | Pass.-Goods |
| o0000—0000o | 2882 | Mallet | Heavy goods |

(Many other wheel arrangements in the last class)

Of the seventeen types above listed there are none in China of the following classes; 080,0100, 2102, 480, or 483. The 442 is used only on the Shanghai Nanking (5 engines). The Mallet is used only on the Peking Suiyan (about 17 engines); The consolidation on the Peking Mukden, Peking-Suiyan, and Peking Hankow (30 engines.) This leaves nine types predominating. What types are in use on the lines with which you are familiar? Tank locomotives are those in which the coal and water supply is carried on the same frame as the boiler and machinery. Wheel arrangements are the same, except that trailing trucks are commonly used. The following types of tank engines are in use in China: 460 (or 064) 262, 040, 060, 242.

Foreign capital invested in the developement of the Chinese railways has had the effect of bringing into this country peculiarities of construction of the nationality of the capitalists with little regard to suitability to local conditions. The present tendency seems to be in the direction of a characteristic Chinese style, embodying in general American construction and British finish.

V. REFERENCES : Locomotive reference material is extremely limited. "Railway Age" and "Railway Mechanical Engineer" are the best American magazines. "Locomotive" and "Railway Engineer" best British. "Locomotive Cyclopedia"(Simmons Boardman, N. Y.) "Railway Operation and Train Control" (Woods, McGraw Hill.) "Loco. of Today" (Locomotive Pub. Co. London)

## LOCOMOTIVE CONSTRUCTION DETAILS

VI. THE BOILER. Locomotive boilers are classified according to the shape of the shell, and the shape and width of the firebox. The shell is designed as a thin cylinder. The longitudinal seam cannot be made as strong as the original plate, and this fact requires a modification of the formula. It is to be noted in all such cases the total strength is no greater than twice the strength of the weak side of the structure. Calling the efficiency of the seam E, the thickness of the plate T, the inside diameter of the boiler D, the working pressure in pounds per square inch p, the bursting load on a inch long is Dp, and the carrying capacity of this inch section one section is twice the strength of the weaker side or 2TEU where U is the ultimate tensile stength of the metal. U.S. law provides that the minimum factor of safety must be 4.5, and the standard strength for boiler steel is 55,000 lb. Hence

$$\frac{2TEU}{Dp} = \frac{110000TE}{Dp} = \text{(or is greater than) } 4.5$$

Hence the minimum thickness is $4.5Dp/110000E$

Note also the convenient rule that if E=82% and p=200. the minimum thickness is .01D. (Check this.) Boiler pressures range from 160 to 220 lb. (180 lb. common in this country). Seams vary in efficiency according to their construction. For sextuple riveting the values range from 81 to 86%.

VII. BOILER SEAMS. Review the principles governing riveted joints from previous work (Mechanics, Strength of Materials and Machine Design.) The most convenient method of calculating efficiencies is that of placing the value of the unpunched plate at 100% = U × T × the pitch of a full section or element of the seam, then evaluating in percentage the various portions at which breakage may occur, and making the appropriate subtractions and additions to the 100%. An important point to note is that the damter of the hole in the plate is 1/16″ greater than the rivet used, and that if properly driven the rivet fills the hole though the nominal size is always given in specifying the seam. The most probable methods of failure in a seam may be listed as follows ;

Tearing between the outer row of rivets.
Shearing all of the rivets.
Tearing between the second row and shearing the outer row of-rivets.
Crushing all of the rivets.
Staggered tearing of the plate between the rows.
Tearing the cover plates.
Taring out the inner margin of the plate and shearing the outer rows.
The order obviously depends on the proportion of the various parts.

VIII. STAYS FOR FLAT SURFACES. All flat surfaces must be stayed against bulging or collapsing as the case may be. The general rule for determining the stress in any flat—surface stay is as follows :

S = (gross area supported less smallest superficial area of brace) multiplied by unit pressure divided by the smallest area of bolt.

Threads in staybolts and rods are usually 12 per inch regardless of diameter. The Whitworth contour is usually used. Stay bolts are frequently made with either rolled or drilled hole at the center to show breakage plainly. This hole is commonly 3/16″ diameter. Iron stays are usually stressed to about 7500 lb. per square inch. (Copper 4500 lb., brass 5,000 lb.)

Tubesheets are usually sufficiently stayed by the tubes themselves within the tube area. A certain number of the tubes are "beaded over" to give them holding power. In figuring the strength of such braces as the diagonal braces sometimes used on front tubesheet and backhead all parts of the braces must be carefully checked for strength and efficiency.

IX. THE FRAMES. The frames of a locomotive are made either of steel castings (forgings when very small) or of rolled plates from 1″ to $1\frac{1}{4}$″ thick. Where castings are used the width of the finished frame is made about .2 of the diameter of the cylinder. Frame strength calculations are purely empirical as the actual stresses can only be determined by excessively cumbersome mathmatical processes. The depth may be determined as follows: (in every case gross depth is meant, before boring for bolts)

At the cylinder fit, D=Piston Thrust/1800 x Width
Over the driving boxes, .666D
Top Rail of bar frame .5625 D
Bottom rail of bar frame .4D

Where plate frames are used existing practice is the best guide. One rule which is commonly used is that the net depth of the frame must nowhere be less than the diameter of the cylinder. This is for the sections ahead of the main driving wheel. Back of the main wheel the depth may be reduced as much as 25% if necessary. The equivalent depth of the "horn—block" should be counted as part of the depth. The depth required over the pedestal may also be calculated by using the stress of 3,000 lb. per square inch, the section considered also including the sectional area of the horn—block.

Bar frames are always braced with castings, while plate frames may be braced by plate braces or castings or both. The following braces usually used should be noted: at the front of the engine the bumper and the bracket holding it in place serve as a cross brace. The cylinder castings, the crossties carrying the front and back ends of the fire box and the heavy block or plate structure at the back end usually known as the footplate furnish the requisite strength. Between the front furnace bearer and the cylinders there are sometimes several braces, depending on the distance. A brace is usually necessary at the back end of the guide bars, and usually there is one placed near the main pedestal.

X. THE CYLINDERS. The cylinders of a simple locomotive range from 15 to 22″ in diameter (much larger in American practice) and the stroke is from 20 to 40% greater than the diameter. (Symbols always used: d=diameter, l=stroke.) The plain D slide valve is used for saturated steam, but the piston valve is virtually a necessity for superheated steam. The cylinder castings may be one, two, three, or four in number, three being usual for plate and two for bar frames. (Make sketches to show arrangment.) Three and four—cylinder simple engines have been used and are growing in popularity. (Not yet introduced into China.)

XI. THE CRANK PINS. The main pin consists of four parts: wheel fit, journal for side rods, journal for connecting rod, and extension for carrying the return crank. The main pin is made appromimately " square " (that is diameter and length nearly equal) and the projected area msut be sufficient to keep the bearing pressure within 1600—1700 lb. The side rod journal is made slightly larger in diameter and about three fourths the length of the main rod journal: a safe bearing pressure is 2000 lb., based upon the largest force which the main side connection can be required to transmit. (The worst conditions occur with broken rods on the opposite side of the engine: the greatest load is about .65p in four-coupled types, .5p for six-coupled engines with main wheel in the middle, and .75 for eight-driver engines and six-driver engines with the main wheel at the front or back. (Explain by means of diagrams.) The pin as a whole must carry the piston load as a cantilever beam: stress limit about 18,000lb.

The other crank pins are designed form empirical fromulae and clearancec onditions. In the case of the front pin, it must often pass behind the crosshead (minimum clearance 1″) and all of the pins ahead of the main wheel must pass behind the connecting rod. The back pin is made identical with the front if possible, and intermediate pins are generally made slightly longer (though of the same diameter) in order to equalize and reduce the offsets in the rods. The projected area of each pin should be checked against a permitted load of 2000 lb. per square inch.

XII. THE AXLES. The weight of the engine produces so small a portion of the stress in the driving axle that it is commonly ignored in making the calculations. The principal stresses are those due to the torsion and bending actions of the piston thrust. The main axle is usually calculated by the use of the standard formula for combined torsion and bending moment: the *equivalent bending arm* is equal to the average of the bending arm and the hypothenuse of torque arm and bending arm. (Fig. 2) The bending arm in this case is the "overhang" (half the difference between the frame center distance and the cylinder center distance) and the torque arm is half of the stroke of the engine. Figured in this way a stress of 23,000 to 25,000 lb. may be used, but the axle must be large enough when new so that from 1/4 to 3/8″ may be turned off in repairing without exceeding the stress limit imposed.

XIII. THE CROSS—CENTER DIAGRAM. (See Fig. 2) The following figures are useful for reference: (all apply to 56.5″ gage engines)

Usual distance between hubs 55″
Usual distance between tires 53 1/4 to 53 1/2″
Usual hub thicknesses 6 to 8″
Usual Journal Lengths 9 to 11″
Usual crank pin lengths 7 to 8″
Bar frames are from 3 1/2 to 5 1/2″ wide.
Distance between centers of bar framess 44 to 46″
Distance between plate frames 47 1/2 to 49″
Thickness of plate frames 1 1/8″ to 1 1/4″

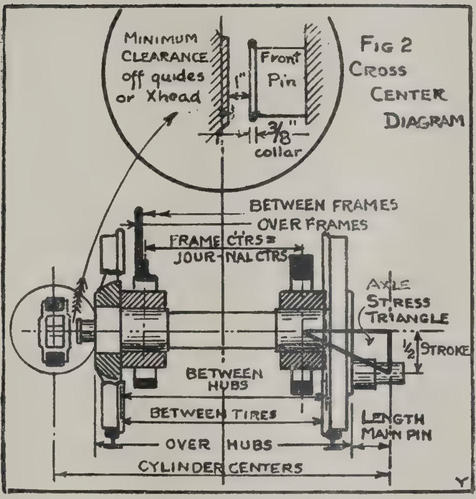


FIG 2 CROSS CENTER DIAGRAM

of support the maximom bending moment. An engine with a two-wheel leading truck repuires a long guide with the support at the extreme end ; a four—wheel leading truck engine can have the bearer more conveniently placed and the guide made shorter. Provide over—travel and wipe—grooves. The guide is figured of a simple beam and the maximum stress allowed 8,000 to 8,500 lb. For a middle-supported guide the depth should not be less than 2 3/4″, regardless of width or stress. Check the ends of the guides for clearance off of the top and bottom of the rod body in extreme angular positions.

XVIII. THE GUIDEBEARER. The bearer is made either of a steel casting or a piece of inch or inch—and—a—quarter steel plate. The form and shape are determined by the kind of frame used and the load carried. Check for clearance off of the link tail, the counterbalance on the nearest driver. the radius rod, and the tires of the engine truck on the sharpest curve. (This requires usually about 3 1/2" clearance. The stress is the sum of the direct load and the bending stress induced by the direct load. Limiting stresses :

For steel casting 6500 lb. (stresses should not vary more than ten percent. on the two sides, requiring an eccentric section)

For steel plate, 14,000lb.

If the bearer is attached to a guide yoke check the bolts for shear.

XIX. THE CONNECTING ROD. The front stub is usually made solid with adjustable bearing. The width is determined by the length of the wrist pin. The brass does not usually extend beyond the steel portion. Stresses : tension 7,000 lb., with the oil cup side slightly stronger. Bending in end of stub 24,000 lb. based on Pl/8.

The body of the rod is made either rectangular or I section, with a slight taper toward the front in the vertical dimensions. If P is over 20,000 lb. or L greater than 60″, use an I section. The rod body is deigned for the following conditions :

a. Tension at smallest area 8000 lb.
b. Column stress, lateral bending as column with fixed ends.
c. Some at high speed with half piston thrust.
d. Column stress, bending vertically as column with pin ends.
e. Same at high speeds with half piston thrust.
f. Stress due to cenfaifugal force.

The column formula most used is the modified Ritter formula:

$$Sc = \frac{P}{A}\left(1 + b\frac{L^2}{r_g^2}\right)$$

The following limitations are placed on this formula: the slenderness ratio ($L/r_g$) must not be greater than 120; the combined stress must not be greater than twice the direct stress.

The general formula for centrifugal stress in a steel rod, revolving at both ends is

$$\frac{r \times h \times N^2 \times}{2\,000\,000}\left(\frac{L}{r_g}\right) = \frac{r.\,h.\,L^2}{18 r_g^2}$$ when N=336 R. P. M. (note this is the speed at which V in m. p. h.=diam. drivers in inches).
For the connecting rod, divide by 2.
For any other speed multiply the result by the square of the speed ratio. In the formula r is the radius of rotation in inches (=half stroke) and h is the depth of the rod section. The formula is applicable te any section of body.
Tee fcllowing stress limits are representative:

No. b or c, 13,500 lb. No. f, 10,000 lb. No. e plus f, 13, 500 lb.
Passenger engines should be designed for operation at V = D speed, while goods engines can be safely figured for 300 R. P. M. and shunting engines for 250 R.P.M.

The back stub of the rod has its size determined by the size of The crank pin, already settled. The steel portion is about .7 to .8 as wide as the length of the pin. The solid or box type stub, the strap end and the fork end (open at the back) are used, with the U bolt type also favored in Europe. The following stresses may be used:

**a.** Tension in net section of strap 7000 lb., with the total area where oil-holes, strap bolts or wedge-bolts pass through slightly greater.

b. Shear of bolts, key or filler blocks 9,000 lb.

c. Flexure at back of strap based on Pl/8, 24,000 lb.

(In case of a solid or box type stub, the stress should be 20,000)

XX. THE SIDE RODS. Side rods are made with the solid and strap type stubs, and also with the non—adjustable "eye" type, which is the most common. The bodies of the rods are made I section or rectangular, the I section being preferred for rods of greater than 60″ length. They are not generally tapered, though in the case of very long rods as on a 4-4-0 it is sometimes advisable to use a fish-belly shape—deepest in the center. The first step in the design of the side rod is to draw an alignment diagram, showing the positions of the hubs, and the lengths of the various crank pins. The relation between the various rods, the bodies, stubs and brasses must be studied carefully in order to reduce the lateral bending stresses due to offsets to a minimum. In figuring offsets, only the steel portion of the stub needs to be considered, though the steel may not be (and generally *is* not) central with the mid—length of the pin. The loads on the various portions of the rods are determined on the basis of the largest number of wheels which may be slipped, in either normal operation or with any combination of broken rods on the opposite side. For example, a four driver—engine might have one side rod removed: in this case the opposite rod would be required to deliver two thirds of the total piston thrust. In the same way the load on the side rods of a six—coupled engine (main wheel in the middle) is half of the total piston thrust, and in an eight—coupled engine is about three fourths of the thrust on the middle rods and three eighths on the end rods.

Stresses and conditions:

Stubs: when stubs of the apjustable type are used, same design data as for connecting rod may be used. For eye-type stubs the pressing in of the bushing usually imposes a heavier stress than the piston thrust, and the following represents good practice: thickness of bushing 1/6 × diameter of the pin, but never less than 3/4″; minimum thickness of metal a round bushing 1.5 × thickness of bushing. Steel portion of stub from .7 to .9x length of pin. The net tension in the steel based on the proper proportion of the piston thrust should not be greater than 5,000 lb. The strongest portion of the stub must be through the oil cup.

Knuckle pins: bearing pressure 5500 lb. pes square inch. Tension in joint not greater than 5500 lb. at any point. The upper part of the tongue is usually the weakest place. A bushing of steel tubing is usually used around the pin.

Rod Bodies: Sections having the same load are usually made the same dimension, regardless of length, hence calcutions must be based on longest rod. Do not fail to include offset stress, adding this to the "column lateral" stress. The same stresses as are used for the connecting rod are to be used, the loads reduced to the proper proportion, for which note above and also under crank—pins.

## SCHEDULE OF PRELIMINARY DESIGN CALCULATIONS

1. Type············Axle weight limit ············
2. Cylinders············Driving wheels······ ···Pressure·········· T.E.··················
3. Boiler Diameter············at············Type················No. Couoses············Size of firebox············Area.············No. tubes and flues··············Length tubes··············Diameter················ Spacing····················
4. Sketch of seam to be used.
5. Efficiency through outer row··········Second row············ Third row············Shearing of all rivets···············Tearing inner margin········Crushing all rivets···········Other weak points········

6. Factor of safety specified·········Strength of plate·········

First course : I. D···········Load········Thickness········Cap'y·······
Factor·················

2nd „ I. D·······Load········Thickness········Cap'y ···········
Factor··················

3rd „ I. D········ Load········Thickness·········Cap'y·········
Factor··················

7. Thickness of other sheets : Smokebox·········Liner·········
Front tube sheet········Throat or hip·········Roof········ Sides·········
Crown···········Sides········Backhead········Door sheet········Dome
type···········Mudring Corners···········Front········Sides···········
Back·················

8. Crown stays, material········type········diameter··········
minimum area·············superficial area···········Spacing···········
Load···········Stress··············

9. Water space stays, material················Type···············
diamerer············minimum area···········superficial area···········
spacing···········Load···········Stress·······················

10. Tube sheet bracing········style········area per brace·······
load per brace···········stress at weakest point in brace··············

11. Backhead bracing········style········area per brace·········
Load per brace··············stress at weakest point·······················

12. Main crank pin : P············Bearing Pressure···············
Area········Dimensions········Main side connection : Load···········
Bearing Pressure········Minimum area·········Dimensions··········
Bending moment (ignoring side rod effect) ······ ··Z·········S·········

13. Other Crank pins : Intermediate···········Front···········
Back··················

14. Frame type···········width or thickness···········If plate,
make sketch of horn—block. Area equivalent to········″ depth of
frame plate. Necessary depth of frame over main pedestal·········
Bar frame : Depth cylinder fit··············Over Pedestal·············
Top rail···········Bottom rail··············

Distance between frames············Distance over frames··············· Distance between centers of pedestal faces···············(Y)

15. Distance between cylinder centers·········(C) Overhang··· ············(Q) Length crank pin (=distance from face of hub to mid-length of main pin)·········Thickness of driving wheel hub············ Distance between hubs········Length of journal (=distance between hubs minus distance Y)·························

16. Main axle : Half stroke·········(l/2), Y·········Hypothenuse of l/2and Y=·········(H) Average of H and Q=equivalent bending arm·········Equivalent bending moment······S·········Z·········D·······

17. Axle load x .35=···········(load on one journal) Projected area of main journal as figured············Bearing pressure··············

18. Other journals. Axle load x .40=················Necessary projected area·········Diameter required···············

19. Cross center summary : Cylinder centers············Length crank-pin············Thickness of hub···········Between Hubs··········· Length Journal···············Journal centers···············(=center of pedestal face=frame center for bar frame)

20. Other Crank pins. Does front pin pass guide?············ Crosshead?···············Wrist pin?···············Distance from outside of hub to center of cylinder : ·········Trial width of guide············ Of crosshead············Possible length of pin allowing 1" clearance and 3/8" collar········· ; Load on pin·········Area required············ Diameter required··············Size intermediate pins··················

21. Knuckle pins. Load : ·········Area required for bearing ·········Trial size of pin············Area required for tension············ O.D. Tongue and jaw·····················

22. Guides. Number·········Width·········Diagram of support ············Load on guide : (L=·········l =·········offset of cylinder center············″ above wheel center) Normal pressure on guide, maximum············Maximum bending moment············S············Z Depth··············

23. Guide Bearer : material·········inside or outside············ Maximum width of section·········Trial section : I·········Zt·············

Zc·········Flexural stress············Bending stress············Combined Stresses···············

24. Piston Rod Length, centers: ···············Trial diameter of body··········Colmmn stress··········Diameter piston fit thread········· No. threads perinch············Type thread·········Diameter at root Area as root············Tension at root·········Allowable shear in key ············Required area ············size key·········Largest diamster of crosshead fit············diameter at of minimum area·········minimum area············Stress···············

25. Crosshead. Length of wrist pin·········Area required for bearing·········Diameter············Check for shear············Necessary area of neck or barrel·········O.D·········Hoop·········Normal pressure on guide············Area shoe············Width············Length··········

26. Connecting Rod. Size crosshead pin···············Crank pin············Length············Front stub type············Depth below bearing············Thickness at end, M= ············(Base on P l/8 and S = 24,000)=·········h=············Back stub, style············Area for bolts or fillers in shear············Number and diameter··············· (S=9000 lb.) width of steel············Thickness at back end ········· Depth at oilcup······at wedge bolt···············at strap bolt ············ Body : Trial sections : I, A: $r_g$ around X and Y axes.

a. Column stress starting, vertical·······························

b. Column stress starting, lateral································

c. Centrifugal s ress··················

d. Direct stress at smallest area··············

27. Side Rods. (For knuckle pins and joints see above.) Sizes of pins·········Main side eye : Width steel (= about .7 length of pin) ·········Thickness of bushing·········minumum metal around bushing ··············Tensile stress············ ··required depth if greater than minimum··············Depth under oil cup·····················Front side rod stub ; bushing············minimum················necessary············ Back side rod stub : bushing·········minimum······necessary········· Intermediate stub : bushing············minimum······necessary·········

Rod bodies : figure same as connecting rod, using proper loading, and same stress limits, lateral stresses to include offset flexure.
Alignment diagram :··················Trial sections·························
Offset Front rod :············Intermediate···············Back···············
Bending Moment, Front rod·········Intermediate·········Back·········
Z, Front rod···············Intermediate···············Back··················
Offset flexure Front rod·········intermediaie·········Back···············
Front rod, stress (a) plus offset flexure·········stress (c)·················
(note that side rod revolves at both ends, hence denominater is 100) Stress (c) plus (a)/2·········(Similarly for other rods.)

## THE LOCOMOTIVE AS MACHINE

XX. PROGRESSION. Thre progression of the locomotive is caused by the pressure alternately of the steam on the front and back cylinder heads, and of the axles alternately against the frontland back of the journal. All other forces are balanced by reactions and have only straining effects within the machine itself.

The locomotive invariably runs UNDER when running forward. Hence, when the crank pin is above the center line of the axle, the steam pressure is on the back cylinder head, and vice versa. In the first case, the pressure of the steam on the cylinder head is P; the pressue of the axle against the box is *greater* than P: if the vertical distance of the crank pin above the center of the wheel is salled V, then $V = 1/2 \sin \theta$, and the force pressing against the side of the journal box is

$$\frac{P\,(D/2 + V)}{D/2}$$

(1 = stoke
(D = diamter of driver
(θ = crank angle

and since the forees producing progression work in opposite directions, the net force causing the engine to move forward, ignoring friction, is

$$-P + \frac{P\,(D/2 + V)}{D/2} = \frac{Pl \sin \theta}{2}$$

When the pin is below the center of the axle,

$$+P - \frac{P\,(D/2 - V)}{D/2} = \frac{Pl \sin \theta}{2}$$

The two expressions are identical. The average value of sin $\theta$ for the complete travel of the crosshead in one direction is $2/\pi$, and since P may be written (for slow speeds only) $pd^2\pi/4$ where p is the boiler pressure and d the diameter of the cylinder,

$$\frac{Pl \sin \theta}{D} = \frac{p.l.d.^2\pi.2}{D.\pi.4}$$

and since there are generallyt wo cylinders, $pld^2/D$. This formula gives the "tractive effort," the pulling force exerted at the drawbar. The formula is invariably written with a multiplying factor to take into account the reduction in force due to friction, insufficient steam capacity and wire-drawing as the speed is increased. The same formula may be derived by another method. The work at the drawbar through one revolution of the wheels, if T is the average force exerted, is $T \times D\pi$. The work in the cylinders per stroke is PAL, and hence for a complete revolution, $p\,A\,l \times 4 = p\,d^2\pi l = 2\,T\,D\,\pi$ and since there are two cylinders, $T = P\,l\,d^2/D$ as before.

XXII. THE FACTOR TO BE USED. The pressure P represents the average mean effective pressure through a cycle. At low speeds where steam is admitted for nearly full stroke, the statement above ($P = pA$ where p is the boiler pressure) is approximately true, and the reasons for departing from this fact have been suggested. The formula as written is called the theoretical tractive effort (To for convenience). For any given operating conditions the available tractive effort is k To. The evaluation of the constant "k" will be studied later.

It will be evident that in order to obtain tractive effort at the drawbar there must be no slipping of the wheels upon the rail. The coefficient of friction for rolling is from .2 to .3, with .22 usually accepted as a conservative mean value for locomotive operation under good rail conditions. The best conditions are a very wlel washed rail (as in a heavy rain) or a very dry clean rail. Grease, frost, snow, and dust greatly reduce the coefficient of friction. From the nature of the case it is evident that Wf (driver weight times coeff.) must be greater than the tractive effort. The ratio $W/.85T_0$ is commonly called the "factor of adhesion."

XXIII. HORSEPOWER. The horsepower exerted by a locomotive is a function of the tractive effort and the speed. If the speed is V m.p.h. = V x 1.466 ft. per second (check this) the expression for horsepower is

$$HP = \frac{T \times V \times 1.466 \times 60}{33{,}000} = TV/375$$

The question of traction and resistance is considered in sections L-LVI.

XXIV. CENTRIFUGAL FORCE ON CURVES. Centrifugal force plays a vital part in locomotive operation. First, the effect of the force on the engine as a whole when rounding a corve should be examined. If in the general formula for centrifugal force are substituted the values W/32 (for M) and 1.466 V for v we have

$$C = \frac{Mv^2}{r} = \frac{W \times (1.466V)^2}{32r} = \frac{.0675\, W\, V^2}{r}$$

The radius of curvature is expressed variously: by British engineers in chain (1 chain = ? yards), by French engineers in meters, and by American builders in degrees. A 1 degree curve has a radius of 5730 feet (this value selected since a 100-foot chord subtends one degree at this radius), and hence for any curve its degree is 5730/R where R is the radius in feet.

The action of the centrifugal force tends to tip the engine over. The resultant of the weight of the engine and the centrifugal force should act a safe distance inside of the rail, and very serious accidents are on record as a result of excessive speed on curves. The track may be made safer by elevating the outer rail, which result in all of the forces being revolved in the direction of safety by the amount of the "superelevation angle". The highest superelevation is about 4″. Obvionsly for any locomotive or car with a given height of the center of gravity (which is approximately at the bottom of the boiler in the case of a locomotive) the superelevation is correct for only one speed. (See Fig.3)

XXV. CENTRIFUGAL FORCE WITHIN THE LOCOMOTIVE. The action of centrifugal force and its related force, inertia, gives the locomotive engineer many difficult problems. In this case several reductions may be made which put the usual formula into very convenient form for use. When the number of R. P. M. is 336, (that is $V=D$)

$C=1.6Wl$ wherc W is the weight of the revolving part
l is the stroke in inches.
(PROVE THIS.)

For any other speed, centrifugal forcc my be found by the principle that it varies with square of the speed.

XXVI. INERTIA. Closely related to centrifugal force is the horizontal inertia of the reciprocating parts. The formula for the acceleration of the reciprocating parts as developed in Mechanics texts is

$$a = \frac{v^2}{r}\left(\cos\theta + \frac{r}{L}\cos 2\theta\right) \quad \begin{cases} r = \text{half stroke} \\ L = \text{length of connecting rod} \\ \theta = \text{crank angle} \\ v = \text{rotational speed} \end{cases}$$

(all in f.p.s. units)

The force required to produce this acceleration (hence the force alternately imparted to and taken from the machine as a whole) is

$$F = 1.6\,Wl\left(\cos\theta + \frac{r}{L}\cos 2\theta\right)$$

Note that the expression is merely the centrifngal force of a corresponding weight, multiplied by the cosine to give the horizontal component, and the result modified by a "warping factor" taking the angularity into account. This is always a purely horizontal force, the parts having pure reciprocation having obviously no vertical movement and hence no possible vertical accelecation. The parts which rotate have the latter, and the connecting rod, which has an intermediate motion will also have an intermediate value for its inertia force.

Horizontal inertia of pure reciprocating mass, $C\left(\cos\theta + \frac{r}{L}\cos 2\theta\right)$

Horizontal component of inertia of rotating mass $C'(\cos\theta)$

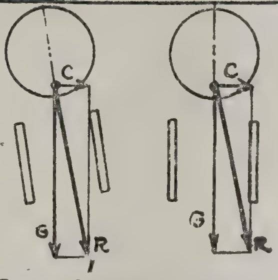


FIG. 3 SUPERELEVATION.

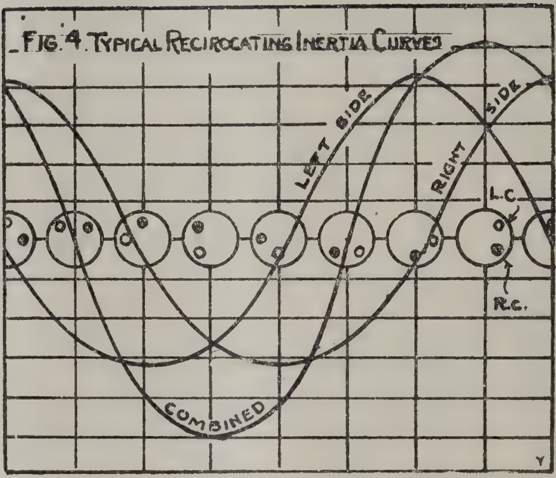


FIG. 4. TYPICAL RECIROCATING INERTIA CURVES

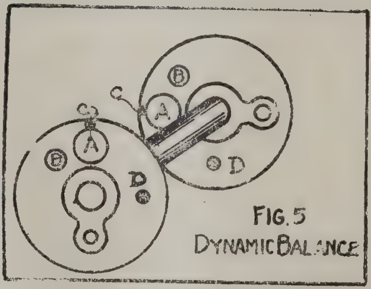


FIG. 5
DYNAMIC BALANCE

Hence, if the weight of the connencting rod be considered as concentrated at its center of gravity, a distance from the large end x, the warping component becomes

$$r\frac{x}{L^2}(\cos 2\theta)$$

and the horizontal inertia is $C''(\cos\theta + \frac{rx}{L}\cos 2\theta)$

See Fig. 4 for inertia effect curres.

XXVII. COUNTERBALANCING. The moving parts on one side of a locomotive (neglecting the parts of the wheels which automatically balance themselves) amount to from 1 to 1.6% of the total weight in working order. Hence at 336 R.P.M. the inertia forces on one side (using the larger percentage) will amount to about W1/35, and the combined effect of both will be W1/25, which is practiclly W, the weight of the engine. Passenger locomotives at least should be safe at this speed, and no machine could endure the action of such forces. However, it is possible to compensate for so large a proportion of this inertia that the remainder has a relatively negligible effect on the motion of the engine.

To counterbalance the rotating parts it is necessary to place in the same wheel and IN THE SAME PLANE a force of equal and opposite centrifugal force which in this case is merely one of equal and opposite centrifugal force : equal and opposite moment about the center of the axle, since the speed of rotation is the same for the weight and for the balance. The "same plane" specification is manifestly impossible, and if *not* in the same plane, balance and weight will have a couple set up between their respective centrifugal forces. Employing the principles of dynamic balance, however, it is possible to place in the wheel on the opposite side of the engine a weight wich will compensate for this couple, and the weight of which in turn must be compensated for in static or moment balance. Very often the transverse moments are neglected as will be seen later.

The reciprocating inertia could be compensated for by the use of a set of dummy parts of the same weight and exactly opposite motion, but this is also impracticable. The only feasible method is to place in the wheels as partial compensation an additional rotating balance, the horizontal component of which is nearly epual to the inertia of the reciprocating parts. Thus balanced another difficutly is encountered : this rotating weight has a vertical component which is entirely unbalanced. This phase of the problem will be discussed under the heading "Dynamic Augment".

There is no possible method for obtaining exact balance for the connecting rod,and the best solution seems the division of the weight between the two ends considering one portion as reciprocating and one part as rotating, adding weights to the reciprocating and rotating balances accordingly. The method of division varies greatly with different builders and is a much argued question. A half-and-half division is employed more often than any other, and is probably reasonably accurate where the body of the rod is heavy relative to the ends as in the case where the piston thrust in thousands of pounds is not greater than about 2/3 the length of the rod.

XXVIII. PRACTICAL METHODS OF COUNTERBALANCING. The first point to be settled is the method of balancing to be used. In America each side of the locomotive is balanced independently of the other, with no referenee to the lateral disturbances. It is evident that this method is reasonably satisfactory, for in America the counterbalance is put to the severest test, and it is applied to some 70,000 engines. On the continent of Europe the attempt is invariably made to care for the lateral forces and obtain as nearly as possible perfect dynamic balance. British builders are divided in opinion and practice, and usually use a compromise method. The former method is commonly called "straight" balancing and the latter "cross" balancing. The commonest form of compromise balance is to balance the rotating weights without reference to the opposite side or to any lateral disturbance, but to treat the reciprocating weights of both sides at the same time. This results in the reciprocating balance being placed 45 degrees away from the main balance in theory; in fact the two are combined and the resultant placed as called for by the values of the components, and the position will be some degrees back of the position opposite the crank in the direction of rotation.

XXIX. DYNAMIC AUGMENT. The second question to be settled is the amount of vertical disturbance which is permissible. This vertical disturbance tends alternately to increase and decrease the static weight upon the track, and cases have been known of a wheel lifting clear of the rail and of rails "kinked" as a result of the force of the descending balance. As both lifting and hammering effect vary with the spuare of the speed the augment allowable is generally expressed as a certain percentage of the weight on the wheel or axle at a given speed. This percentage is fixed in some countries by law, and by most railways by engineering department rules. For a single wheel, if it contains an unbalanced weight W the D.A. at 336 RPM is evidently 1.6 W l. Suppose this is K % of the weight on this wheel. The worst condition of D.A. comes when both cranks are on the 45 degree points above the center of the axle (raising effect) or below the center (hammering effect) and the total force is evidently 1.414 times the maximum for one side. However since this force is applicable to two wheels, the D.A. percentage is .707K. Thus after a D.A. limit is stated it has no meaning unless whether applicable to the wheel or to the pair of wheels is known. Twenty percent. D. A. per AXLE at 40 miles per hour may be considered as a representative value for the Chinese Government Railways.

If no attempt is made to care for lateral disturbances the reciprocating weight is placed with the rotating weight opposite the crank. If dynamic balance is desired, it must be remembered that the two reciprocating inertias on the opposite sides of the engine form a couple of continually varying magnitude, the mean force to counteract the couple action being a force of 1.414 times the value of the largest force on either side and the meanposition being the center line of the engine. Hence a "bob" balance might be keyed to the main axle having the required centrifugal force, but in practice the bob is divided in two parts and one half placed in each wheel in the proper angular position. This permits the use of much less dead weight (the arm being greater for the wheel position, the weight may be less) and also eliminates the many practical objections to the bob itself.

The reciprocating counterbalance problem thus becomes one of placing sufficient reciprocating compensation in the wheel to reduce the inertia effects to such a value that the motion or the riding of the engine will not be sensibly affected, while at the same time reducing the amount to such an extent that the vertical effects will not be objectionable. Obviously the reciprocating compensation may be divided up among all of the driving wheels. It has been found that in general if about one half of the reciprocating parts are balanced, the inertia effects will not be injurious, and this amount can generally be obtained without exceeding reasonable D.A. limits. It is to be noted however that the D.A. limit *must not be exceeded*, even at the expense of apparently inadequate reciprocating balance.

XXX. PROCESS OF MAKING A DYNAMIC BALANCE. Obtain first all the necessary information for the work, including the drawings of all parts concerned (wheels, pins, rods, etc.) and determine the weights of such parts as are needed. Determine the limits of the balance by its clearances off of the rods, spring rigging, guide bearers and other parts, and how far the outer edge of the balance must be kept from the edge of the wheel. Second : Find the proper amount of reciprocating balance as fixed by the D.A. specification and the amount of reciprocating compensation desired. Third : Make a diagram of the main crank pin and the weights distributed along it, and determine the center of gravity of the whole and the relation of the center to the the center of the balance to be used. Fourth : knowing the distance between the center plane of the parts to be balance, and of the balance, and the amount of the parts to be balanced find the amount of weight which must be added to the opposite balance to compensate for the moment. (Note that to secure static balance the amount of this "counter—counterbalance" must be added to the main balance weight. Fifth: diagram the position of the three balances in the wheel. These are as follows: balance A opposite the crank, equal in moment to the weiget of the revolving parts; balance B, the reciprocating balance 135 degrees away from the crank so placed that it bisects the 270 degrees angle between the cranks; balance C, opposite the crank, (in phase with A) the addition to the balance to compensate for the dynamic balance on the opposite side; D, the dynamic or counter--counterbalance, in phase with the opposite crank. (Make a cardboard model with the cranks in the proper position——this developes the fact that the two wheels will be relative exactly alike when thus balanced.) (*See* Fig. 5) Sixth: find the resultant value and angular position of the balance.

The method of placing the desired balance in the wheel will be taken up in a succeeding paragraph.

### XXXI. PROCESS OF MAKING A STRAIGHT BALANCE.

First and second as before.
Third: Add together all the rotating weights in the wheel and the predetermined amount of reciprocating weight and place the whole in the (wheel) opposite the crank.

### XXXII. PLACING THE BALANCE IN THE WHEEL.

Since a balance weight of a given moment about the center is desired, it is evident that the further the center of gravity can be placed from the center the less will be the weight required. All moments thus far considered have had the crank radius (half stroke) as the arm. The actual shape of the balance block is either a sector, segment, or lune. The latter is credited with having the best appearance, while the first has the most useful properties and is somewhat the easiest to calculate. The math—matical calculation of any of these figures is rather cumbersome, though all handbooks give the necessary formulae. The simplest method is to cut a profile of the balance from stiff paper, find its area by the use of the planimeter and its center of gravity "swinging it" from one or more points of support.

area by the use of the planimeter and its center of gravity "swinging it" from one or more points of support.

In the case of small wheels and large piston thrusts difficulty may arise in that sufficient metal can not be placed within the limits of the balance. The first recourse is to make the balance block hollow and fill it with molten lead. This increases the actual weight and effective weight as well by from 15 to 30%, according to the method used in coring and filling. (Note that for small adjustments, pockets may be made in the rim of the *wheel*, and filled with lead if desired.) This difficulty arises when 54″ wheels (48″ centers, a common size on heavy goods engines in China) only with piston thrusts in excess of 65,000 lb. A lead filled balance should never be used unless absolutely necessary. There are no objections except the slight additional expenses incurred to a balance which merely has lead pockets.

In general, if the desired combination of reciprocating and rotating balance can not be secured in the main wheel, the deficiency may be charged to the reciprocating weight and distributed among the other wheels, with care that D.A. limits are not exceeded. If the deficiency is so serious as to involve a lack of sufficient rotating weight, only the conditions of the problem can dictate the treatment, rnd an enforced speed limit is a frequent recourse in order to keep the D.A. within the desired limit.

## THE LOCOMOTIVE AS A POWER PLANT.

XXXIII. CHARACTERISTICS. The steam locomotive is a complete power plant, portable and self-containd. The characteristic feature of this plant is the intensiveness of its operation, as is shown by the fact that about 30 times as much horsepower is produced per cubic foot of the plant as in the most compact stationary plant, and even the locomobile and the ordinary steam ship require considerably more space per horsepower. The steam automobile engine and the destroyer are still more economical of space in terms of output, however. The firing rate on the grate of a locomotive is commonly ten to fifteen times as much as for a is made possible by the high induced draft produced by the action of the steam exhausting into the smoke box.

The three functions of the power plant are to generate steam, to turn that steam into mechanical work, and to deliver the energy in useful form. This forms a convenient outline for a division of the study of locomotive performance. The violence of the boiler action has been already mentioned The principal characteristic in which the second part of the process varies from stationary process is in the use of highly superheated steam in simple cylinders (usually) and non-condensing running. Working pressures are higher than in stationary practice. The complications introduced by the compound cylinder are not usually warranted from an economic standpoint. The transmission and delivery of the power are vitally different from power-plant practice, and have already been studied.

In order to improve the performance of existing engines and predict the action of future machines it is necessary that performance figures as given by test results should be carefully studied. The most useful and in fact only accurate form of locomotive test is the "plant-test" in the laboratory. Enough laboratory work of the sort has been done to make possible many useful genernalizations, though much information is still to be obtained. The essentials of the laboratory testing outfit are are the usual set of instruments employed in stationary plant testing, with in addition a set of rollers upon which the locomotive may run, and a means of absorbing the power generated. A dynamometer for measuring the tractive effort delivered is a convenience. A set of prony brakes could be used, one on each of the supporting axles, but an error is introduced thereby. What is it?

XXXIV. FIREBOX PERFORMANCE. The subject of steam generation divides into the matters of generating the heat in the furnace and of absorbing the heat. The efficiency of the firebox veries greatly with the air supply, rate of combustion, and to some extent with the quality as well as the quantity of fuel used. The principal losses which occur in the firebox are the following.

Green coal lost through the greates: approximately constant for any grate and size of coal, hence decreases in proportion with the increase in firing rate.

Thermal loss in the ashes due to incomplete combustion. This loss should be very small in any case and it increases slightly with the rate of firing.

Loss of incompletely burned coal through the front end. This loss depends on the proportions of the firebox, the size of the coal, and for any given conditions almost directly with the draft and combustion. A major loss.

CO loss—due to incomplete combustion and increasing with the rate of combustion.

Radiation: the only radiation *loss* is that dowdward, and through the door. This loss increases with the firebox temperature and is thus greatest whe then firebox is most efficient—evidently a very small loss.

No general law can be laid down as to the variation in firebox efficiency, though it is to be noted that the largest loss (incompletely burned coal in the form of sparks and cinders) is largest at high rates. The point of maximum efficiency represents a proper combination of firebox volume in relation to grate area, character of fuel and air supply which will give the highest temperature for a given firing rate. Best firebox conditions with bituminous coal of good quality are given by a firing rate of about 150 lb. coal per square foot of grate for narrow fireboxes and 100 lb. per square foot for wide fireboxes, these figures representing about two thirds of the ultimate capacity of the grate in each case.

XXXV. HEATING SURFACE EFFICIENCY. The efficiency of the heating surface is measured by the ratio of the heat units absorbed to those offered. The B. T. U. that are not abosrbed are lost at the stack and this is the predominating boiler loss. The percentage of absorbtion depends on the speed of the gases passing over the surface, the condition of the surface as to soot and scale, the temperature difference, and the length of the passage. The second and fourth losses may be eliminated by comparing performances on the same engine at about the same time. The higher the speed of the gases the less time they will be in contact and hence the less the proportion of absorbtion. The speed of the gases is directly dependent on the amount of fuel burned, if the excess air supply is kept proportional. However this loss is mitigated by the fact that as a result of the higher speed the gases at the front end of the tubes are much hotter than they would otherwise be, hence the greater amount of heat absorbed in this portion of the boiler and the higher capacity. It is more difficult to analyze the relation between the heating surface loss and the firebox temperature. The higher the temperature the greater the temperature difference and hence the greater the *amount* of heat absorbed, butthe *proportion* of heat absorbed is not greater unless the firebox temperature can be increased without equally increasing the temperature of the front end gas—this is a question of proper air supply.

XXXVI. BOILER EFFICIENCY. It is the combination of the two efficiencies which is of importance to the engineer much greater than the separated values. In operation, locomotive beilers show efficiencies from 30 up to 80 or even 85 percent. and the rate of combustion is the predominating variable. As the firing rate increases the proportion of loss increases. If the results of a series of tests on the same engine with different firing rates are compared many useful facts can be learned. The first is the relation between the coal fired and the water evaporated. If the efficiency were constant this would be a straight line passing through the origin. However, it is well known that for a given boiler fired with a given grade of coal the efficiency falls off as the firing rate is increased, hence the coal-water curve actually falls further and further below the line representnig the highest possible efficien y, which is attained at the lowest firing rate. The true relation is represented by a curve which eventually becomes a maximum and then falls off, indicating the firing of more fuel than adequate air supply can be obtained for, and an actual decrease in evaporation due to extremely poor combustion. For any given locomotive we are only interested in a certain range of this curve. The lowest limit is the amount of steam produced which will suffice to move the engine itself; the upper limit of firing is the highest rate which can be maintained by the available means of firing,manual or mechanical. The extreme limit of hand firing, which may be maintained for only a few minutes as an emergency measure is about 100 lb. per minute. The ordinary fireman can shovel 3500 lb. of coal per hour for several consecutive hours if the weather is not excessively hot. Mechanical stokers are in use which are capable of handling 12,000 lb. of coal per hour.

XXXVII. EFFICIENCY VS. CAPACITY. In order to facilitate comparison between boilers the coal-water curve just referred to is usually replotted on a slightly different basis: dry coal per square foot of grate per hour (or of heating surface) against equivalent evaporation per square foot of heating surface per hour. The latter is as much a measure of capacity as is the total evaporation and for convenient referrence will be called "Ec". A second line which may be drawn on the same plote has the same abcissae, and as ordinates the equivalent evaporation per pound of dry coal (note, *not* "per hour" !) The latter is obviously a measure of efficiency and will be called "Ee". Finally the most useful relation of all may be defived from the data for the pair of curves just produced--namely the relation between simultaneous values of Ec and Ee—capacity and efficiency.

The Ec-Ee curve is for all practical purposes a straight line. Dr. Goss found that the equation of this curve for the Purdue engine "Schenectady No. 1" using high grade coal was

$$Ee = 12.9 - .36\ Ec$$

Study of recent test material shows that this equation may be used with satisfactory accuracy to represent the performance of engines of modern design with this same graae of coal.

XXXVIII. THE CAPACITY-EFFICIENCY CURVE AS A GENERALLIZATION. A study of the nature of this curve

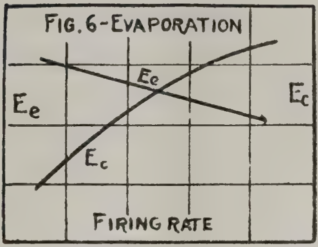


FIG. 6-EVAPORATION

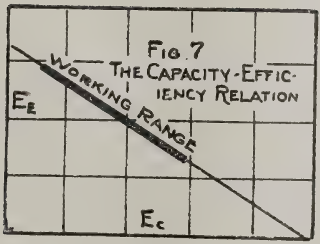


FIG. 7
THE CAPACITY-EFFICIENCY RELATION

shows that it may afford a key to boiler performance in general, for the purposes of checking, comparison, and prediction. Draw the relation in its intercept form as shown in Fig. 7. The heavy portion represents the working range of the engine. Consider first the left end of the curve, as the Ee intercept is approached. The ultimate condition may be visuallized by thinking of a small pile of coal on the grate, producing such a small amount of heat that the smoke box of the engine is practically cold. The only losses are downward radiation, ash losses and green coal lost through the grate—these persist as an inreducible minimum, and the efficiency obtained is an ideal maximum, unattainable under working conditions. Under the ideal ultimate conditions the evaporation (equivalent, of courve) would vary with (not "vary as" as all tests show) the B. T. U. value of the coal, and all results point to the fact that the evaporation improves in slightly greater proportion than a variation in the heating value ; that is, the EFFICIENCY improves with the quality of the coal. This question will be further discussed.

It is much more difficult to speculate on the ultimate condition at the Ec axis, perhaps because it is further removed from the range of operating conditions. The condition may be approximated by the use of a large capacity oil burner or mechanical stoker in connection with a firebox of small size——so great a volume of fuel might be forced through the firebox that the ratio of heat absorbed to BTU available would approach zero. Under these conditions the quality of the fuel is a negligible variable, and the limit is fixed by the size and proportions of the boiler, and mere size is liminated since evaporatien per unit area is the basis of comparion.

XXXIX. EVALUATING THE INTERCEPTS. From the foregoing discussion it becomes evident that a "complete" Ee—Ec curve could be plotted for any boiler fired with any kind of fuel if it were possible to evaluate the intercepts. As has been said comparable tests of boilers with varying qualities of fuel show clearly that the efficiency falls off with the increase in inert matter in the coal. This may be stated in another way: a certain proportion of the thermal content of any fuel is "unavailable", and the evaporative value varies with the "available" portion. (It is worth noting for example that from 700 tp 800 B.T.U. of each pound of coal is required merely to raise the coal to the firebox temperature, and a further amount is required to keep the lunp at the temperature attained.) The total amount of unavailable heat is apparently about 3,000 B.T.U. in coal of any grade, hence the steam value of the coal is proportional to its excess over 3,000 B.T.U. Dr. Goss equation gives an Ee intercept of 12.9 lb., which with the coal of approximately 14,000 B.T.U. used gives an ultimate efficiency of 90%. The Pennsylvania tests furnish ample substantiation for this value. Hence, the ultimate Ee for any other fuel of B.T.U. value B is

$$12.9 \times \frac{B-3000}{14000-3000}$$

and thus knowing the grade of fuel fired one end of the curve may be located. It is also advisable to modify the value of the interept to some extent to allow for the physical condition of the fuel used.

It is doubtful whether the Ec intercept can be located on the basis of boiler proportions from information now available. The data in a general way seem to show that a high value of Ec goes with a boiler of the long tube and relatively small firebox type, but the data is in some cases contradictory and irreconcilable. The great majority of tests give Ec intercepts between 33 and 43—the value from the Goss equation is practically 36. Lacking the necessary data to establish a general law of variation, the value given by Dr. Goss, corresponding with test results from modern locomotives of a wide range of design, is safely conservative for general use. In the use of the Goss equation and coefficients it is necessary to remember that all values are in terms of equivalent evaporation.

For practical purposes the Factor of Evaporation may be taken as 1.20 for a locomotive using saturated steam and 1.30 for superheated steam. ( Check these. ) The heating surface referred to is the fire-side of the firebox, tubes, flues and superheater units, the superheater header and front tube-sheet not being included.

Heating surface calculated in this way will not check with builder's figures: the builders generally calculate on the water side of the tubes and flues and omit the superheater entirely.

XL. THE COLE METHOD. Mr. F. J. Cole has developed a method of estimating the evaporative capacity of a boiler, or rather a figure which represents its relative capacity, in place of a performance estimate. This is summarized in the ALCO Handbook, Woods "Loco. Operation and Train Control" and elsewhere. The method has been quite generally adopted as a method of comparison. The most important figures are as follows:

The commonest combination of tubes, flues and spacing is 3/4″ spacing, 2″ tubes and 5 3/8″ flues. Under these conditions the total evaporation for a single tube and a single flue are as follows:

| Length, feet | Evaporation, One Flue | One Tube |
|---|---|---|
| | (both in pounds actual steam per hour) | |
| 10 | 193.5 | 63.2 |
| 11 | 202.5 | 67.0 |
| 12 | 217.0 | 70.0 |
| 13 | 226.0 | 73.2 |
| 14 | 237.0 | 76.0 |
| 15 | 246.0 | 78.3 |
| 16 | 254.0 | 80.4 |
| 17 | 262.0 | 82.4 |
| 18 | 270.0 | 84.2 |

For other spacings than 3/4″ the evaporation varies practically in proportion to the spacing. For 2 1/4″ tube the evaporation is 16% greater than for 2″ tubes of the same length; for 5 1/2″ flues the evaporation is 2.5% greater than for 5 3/8″ flues of the same length. The evaporation per hour per square foot of firebox heating surface is 55 lb.

After having evaluated a boiler by the Cole figures the firing rate necessary to produce this amount of steam with any kind of coal may be figured by the use of the Goss equation—not for getting the factor of evaporation.

XLI. ENGINE PERFORMANCE. The fundamental difference between the engine performance of the stationary engine and the locomotive is that the latter is expected to work with reasonable efficiency through a range of speeds O to 300 revolutions per minute, and a range of cutoffs from 15 to 90%.

The usual unit of work used in comparing and measuring steam apparatus is the horsepower-hour = 198,000 foot-lb. = 2545 B.T.U. The horsepower-hour may be "indicated" or "drawbar" according to the method of measurement. The efficiency of the engine is inversely proportional to the "water rate" or the actual steam required to produce a horsepower-hour of work. If W is the water rate and Q the heat content of the steam as used, the efficiency of the engine is (2545 x 100÷WQ) per cent. For asuperheater engine Q is approximately 1270 B. T. U., hence the efficiency is practically 200/W. per cent. In terms of the fuel, the efficiency may be expressed as 2545 × 100÷Coal Rate × B. T. U. per lb. coal.

XLII. CYLINDER PERFORMANCE. Cylinder performance is the weak link in all steam power plants, and the locomotive is no exception in this respect. The principal losses are as follows :

Initial Condensation
Incomplete Expansion
Leakage
Radiation

Initial condensation is the result of the temperature variation withn the cylinder, steam at its highest temperature being injected into the cylinder when the latter is coldest. It may be partly overcome by reducing the temperature range within the cylinder—that is, by compounding, but it may be completely overcome by superheating the steam, if sufficient superheat is applied so that the exhaust is slightly superheated. Superheating also increases the specific volume of the steam, hence less weight is required to fill the cylinder. These two advantages are sufficient to give from 30 to 40% better performance than the simple saturated engine.

Compunding has the additional advantage of reducing the incomplete expansion loss, but the compound locomotive is more complicated, hence expensive to maintain. The last two losses are small in well designed and well maintained cylinders. In travelling from the steam dome to the cylinders the steam loses a certain amount of mechanical energy in wiredrawing—this shows itself as a mechanical loss in the initial pressure, and the slight thermal gain (superheat or reducion of moisture content) is incommensurate with the loss.

XLIII. WATER RATE VARIATION. The water rate may be considered as a function of three other variables: admission temperature, piston speed, and cutoff. This may be reduced to a three-variable plot by considering two limiting cases: superheated steam (sufficient superheat so that the exhaust steam is practically dry, requiring about 180 degrees minimum) and saturated steam.

For superheated steam the conditions of variation are as follows: if the speed is constant, shortening the cutoff reduces the incomplete expansion loss ("I. E. Loss") while the increase in wiredrawing at the shorter cutoffs absorbs some of the gain. If the cutoff is kept constant the wiredrawing which increases with the speed is the only loss. In operation, speed increases and cutoff decreases normally at the same time, hence AS THE SPEED INCREASES AND THE ENGINE IS "HOOKED BACK" THE INCOMPLETE EXPANSION LOSS DECREASES, AND THE WIREDRAWING LOSS INCREASES. AT VERY HIGH SPEED (330 to 360 R. P. M.) THE GAIN DUE TO SHORTENING THE CUTOFF IS ANNULLED BY THE WIREDRAWING LOSS, and after this speed the water-rate again increases. It should be noted that operating speeds rarely exceed 250 R. P. M.

FOR SATURATED STEAM the results are greatly different. The wiredrawing loss is larger due to the denser medium; this results in a "critical" speed and cutoff well within the working conditions of the engine at which the water rate is a minimum, and below and above which the water rate increases rapidly, Stated otherwise, the wiredrawing loss outweights the I.E. saving at speeds over 650 feet per minute (piston speed) and cutoffs around 35%. It must also be noted that the general level of the water rate curve for the saturated engine is much higher than that for the superheater engine. The best recorded performance for a simple satuated engine is about 25 lb. for the saturated engine, and slightly under 15 lb. for the superheater.

It must be noted at this point that since the principal saving in steam resulting from compounding and from superheating alike is the reduction of condensation, the combining of the two principles does not produce a commensurate gain in performance. The compound superheated engine gives good performance at relatively low speeds but is specially subject to wiredrawing losses, and at speeds above 1000 feet per minute is probably less economical than the saturated simple engine. (See Fig. 8 for comparative water rate curves.)

XLIV. PERFORMANCE OF BOILER AND ENGINE COMBINED. The relation between the input and output measured thermally (and cylinder output is here considered, eliminating friction) is a combination of the boiler and engine efficiency relations, and for any given engine may be plotted as a relation between speed, horsepower, and firing rate. For comparison purposes the most useful exhibit is a plot with HP as ordinates, speed (usually speed of the piston, but not necessarily if only one engine is being considered or if all under consideration have the same relation between piston speed and speed in miles per hour) as abcissa, and a family of horsepower curves drawn for different firing rates. For a given firing rate (constant rate of steam production) the horsepower curve is the inverse of the water rate curve, that is, H.P.=E/W.(Watch out for tbe factor of evaporation—water rates represent *water = actual steam* while evaporation may be stated either way!) Since each thermal type of engine has a different type of water rate curve (Fig. 8) there is also a characteristic type of horsopower curve for the simple,

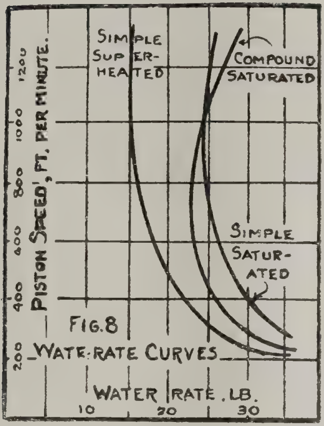


FIG. 8 WATER-RATE CURVES

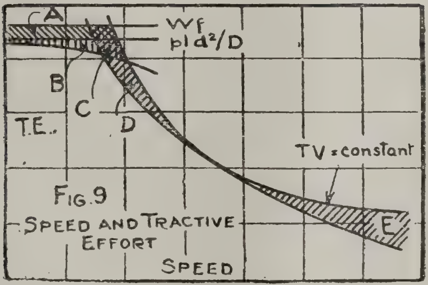


FIG. 9 SPEED AND TRACTIVE EFFORT

compound, saturated and superheated lcomotive. The most important fact about any of these is that at high speeds the value for the simple superheated engine is practically constant.

XLV. THE SPEED—PULL CURVE. From the speed-horsepower curve the most important and useful relation of the series is developed: since speed and tractive effort are factors of horsepower the ordinates can be changed to tractive effort. The speed-tractive effort relation is commonly called the "speed-pull" curve. The nature and geometry of the curve may be analysed as follows: If a constant steam output were maintained and a constant water rate could be used, the resulting horse—power would also be constant, hence the tractive effort curve (plotted against speed) would be an equilateral hyperbola, since $TV = H \times$ a constant. This hyperbola is assymptotic to the vertical axis, but the tractive effort may not go on indefinitely increasing, because the adhesion of the wheels to the rail limit the tractive effort to $Wf$, the weight on the driving wheels multiplied by the coefficient of friction. (See Fig. 9) The true curve for the locomotive deviates considerably from the basic curve. In the first place, the size of the cylinders usually limits the maximum theoretical tractive effort to an amount slightly below the adhesion under normal conditions. This To as has been explained can only be developed at the very lowest speed, and as the speed increases, wiredrawing causes it to fall off. Instead of a constant water rate, the rate is variable with a minimum value in the middle of the working range for the saturated engine and probably about at the upper limit of speed for the superheater. Hence, the speed—pull curve coincides with the ideal hyperbola at one point in the middle of the speed range for the saturated steam engine, and lies below it at every other point; for the superheater, the true curve approaches nearer and nearer as the speed increases, and for speeds above about 900 ft. per minute is practically coincident with the ideal curve.

It is convenient to designate the slow-speed portion of the curve as the "cylinder curve" and the high-speed part as the "boiler line", since the limits are imposed in one case by the size of the cylinder, and in the other by the amount of steam which the boiler is able to supply. Between the cylinder and boiler curves and the broken ideal line it is possible to distinguish several distinct areas, as follow :

A. A space between the adhesion line and the theoretical cylinder line representing the allowance which must be made to insure the adhesion of the wheels under all ordinary conditions of operation.

B. An area between the line just mentioned and the actual pull line, representing wire drawing loss.

C. An area at the "knee" of the combined curve representing the engineman's skill in the use of his reverse lever. If accurately used, notch by notch, the knee would be nearly sharp. If used as usual, that is, hooked back several notches or turns at a time, the curve may not even be a continuous function.

D. An area between the TV=k hyperbola and the actual curve below the critical point of best performance representing cylinder lesses, principally incomplete expansion.

E. An area between the two curves at high speed (practically absent in the superheater case) representing wire-drawing loss.

XLVI. THE PREDICTION OF PERFORMANCE. One of the most important purposes of performance study is prediction by means of generallization of available data. Methods of generalizing the boiler performance have already been presented, and cylinder performance is readily predicted. The results of many tests show that an accurate water rate curve may be assumed on a piston speed base, presenting either what may be considered best possible performance, or usual performance, as desired. The latter is the most useful and the following figures derived from the tables of Mr. F. J. Cole (see A. L. Co. Handbook and elsewhere) carry sufficient allowance for the steam operated auxiliaries.

| Piston Speed ft. per minute | Lb. Actual Steam per I. H. P. H., Superheated | Saturated |
|---|---|---|
| 250 | 34.00 | ......... |
| 300 | 31.00 | 41.20 |
| 400 | 26.70 | 33.30 |

| Pistion Speed ft. per minute | Lb. Actual Steam per I.H.P.H. Superiheated | Saturated |
|---|---|---|
| 500 | 23.90 | 29.70 |
| 600 | 22.60 | 28.30 |
| 800 | 21.04 | 28.00 |
| 1000 | 20.80 | 28.00 |
| 1200 | 20.80 | 28.60 |

**XLVII. CALCULATION OF A COMPLETE SPEED—PULL CURVE.** The process is as follows :

1. Calculate the heating surface, including the fire-side of tubes, flues, superheater pipes and firebox.

2. Calculate the coal factor and the coefficients for the boiler performance equation. (see sections XXXVII and XXXIX.) 3. Calculate the total evaporation for the given firing rate by the following method :

H the heating surface in square feet,

Let E be the desired evaporation, C the firing rate in pounds of coal per hour, and given the boiler performance equation Ee = K— K′ Ec. The equation may be written

$$\frac{E}{H} = K - K' \frac{E}{C}$$

Hence $E = K\,C\,H \,/\, (C + H\,K')$

4. Find the horsepower for each speed by dividing the steam produced by the water rate.

5. Find from the horsepower the tractive effort. It is necessary first to change piston speed to speed in miles per hour, then apply the formula of section XXIII. The final formula may be written

$$T = \frac{21{,}000\ l\ H}{Vp\ \ D} \quad \Big\} \ \text{Vp is piston peed.}$$

CHECK THIS !

6. Draw the wire—drawing portion of the curve from the following data. For an engine with piston valves :

T.E. at Piston speed 0 = .95 × To
T.E. at piston sqeed 100 = .91 × To
T.E. at piston speed 200 = .85 × To
T.E. at piston speed 300 = .77 × To

If the curves for the cylinder and boiler sections intersect, the intersection may be rounded off. If not use the more conservative value, as the smaller value will limit the pulling power.

XLIX. CORECTING THE CURVE TO THE DRAWBAR. The curve as calculated presents the physically non—existent "cylinder tractive effort". The next section takes up in detail the question of traction and resistance but the figures governing the corrections for the resistance of the locomotive and tender are added here for convenience :

Locomotive and Tender Resistance
(pounds per ton weight)

| | Starting | 5 m.p. h. | 25 m.p.h. | 50 m.p.h. |
|---|---|---|---|---|
| Machine frction (Driving wheels) | 35 lb. | 25 lb. | 25 lb. | 25 lb. |
| Inherent resistance, Trucks, tender, | 12 lb. | 5 lb. | 7 lb. | 12 lb. |
| Grade resistance, for each percent., | 20 lb., independent of speed. | | | |

All figures are for convenience of calculation based upon the 2000 lb. ton, which is *not used* in China. The metric tonne and the 2240 lb. ton are the measures used, and ten percent. must be added to the above figures in every case.

## RESISTANCE AND TRACTION

L. TRAIN RESISTANCE. The movement of a wheeled vehicle along the road or rail is analogous to the sliding of a surface over another. Three varieties of resistance are met : the amount required to start the body into motion, the amount required to keep it in motion at constant speed, and the amount required for such incidental qurposes as moving it over local irregularities, altering the speed, etc. In the case of the railway car these resistances are called respectively starting resistance, inherent resistance, and incidental resistance. It is to be noted that either of the first two may be present alone, or combined with the third, but the first two can not be present at the same time, if the train is regarded as a unit. Inherent resisistance may be defined as the resistance to motion on *straight, level* track, at *constant* speed, in *still* air at *moderate* temperatures—note the five qualifications. Conversely; incidental resistance is that caused by speed variation (may be plus or minus), curvature of the track, grade (plus or minus), wind (plus or minus) and extreme temperature, only cold being of considerable effect. Starting resistance is the force which is required to break the adhesion of the sliding surfaces, a force from two to four times as great as the inherent resistance at the lowest speeds.

LI. INHERENT RESISTANCE. The problem of determining the inherent (frequently miscalled "true") resistance was first attacked nearly a hundred years ago, and the usual method of attack has been the attempt to generallize the results of a series of tests of more or less limited scope and application. There are numberless resistance formulae, the general form of these expressions being

$$\text{R (in pounds per ton)} = A + BV + CV^2 + DV^n$$

where A,B,C and D are constants found from the test results, any one of which might be O. Later the attempt was made to express resistance as a function of the weight of the car, that is,

$$\begin{array}{ll} & \text{R (in pounds per ton)} = E - FW^n \\ \text{or} & \text{R (in pounds per car)} = EW - FW \end{array}$$

Neither of these is complete, for obviously the true relation is of the type of these $R = f\ (W, V)$. Consider in detail the components of rolling or inherent resistance. They are rolling friction, flange friction, journal friction, air friction, and oscillation. Consider the variation of each with speed and car weight :

| | For a given ear, that is, Constant weight and variable speed. | For a given speed, that is, Constant speed and variable weight. |
|---|---|---|

| | | |
|---|---|---|
| **Rolling Friction :**<br>$R_f$ = constant as related to speed ;<br>= constant as related to weight. | the car wheel makes a depression in the rail which must be *pushed along* as the car moves. There is no relation between this force and speed apparent. (Do not confuse force and work.) | The amount of deformation varies with the imposed weight, hence the force per unit of imposed weight is constant. |
| **Flange Friction :**<br>$(R_{fe})=k_1+k_2V^n$<br>$(n>0<1)$ | Flange friction is dependent on track conditions, the amount of play between flange and rail, and the shape of the flanges as well as many more involved variables. For a given track and a given set of car running gear : | |
| $=k\pm 0W$ | No known law, but always present; power required varies with V to a power slightly above the first, hence force increases slightly with speed. (Note that the power does not very AS the speed.) | Closely proportional to car weight, hence resistance in pounds per ton constant. |
| **Journal Friction :**<br>$R_j=k_1+k_2V$<br>$=k_1-k_2W^n$ | The total friction or work of friction increases faster than the speed, hence the coefficient of friction varies with the speed. The variation is approximately linear. | The total friction increases with the imposed weight but in a lower ratio, hence the coefficient decreases. |
| **Air Friction :**<br>$R_a=kV^2$<br>$=k_1-k_2W^n$ | Increases with the square of the speed. | Is practically constant for a car of given size and shape regardless of |

| | | |
|---|---|---|
| | | weight, hence coefficent is lower for heavy car. |
| Oscillation :<br>$R^o=k_1+k_2V^2$<br>$+k_3V^n=$<br>$k=OW$ | Varies with the square and higher powers of the speed. | Varies almost directly as the weight. |

From what has been said above it is clear that the general form of the equation for resistance in terms of V is

$$R=A+BV+CV^2+DV^n$$

and in terms of W,

$$R=A-BW-CW^n$$

There is no available information warranting an attempt to separate and evaluate the various coefficents except possibly the air friction and the journal friction in relation to speed, hence all are lumped into the form of the general equation. The theory as proposed checks well with the actually known forms of the curves. The geometrical form of the R—V curve shows that the coefficients of V are all postive and the unknown powers of V may be combined into a single term $V^n$ with an exponent greater than 2, if they are of sufficient magnitude to have any considerable effect on the shape of the curve. The form of the R—W curve indicates a negative coefficient for the first power of W, also for the nth power, and that n is less than 1.

The most useful set of resistance curves are those of Prof. Schmidt, combining the three variables in a single plot. The curves for goods trains are given in Marks (p. 1224). There are also similar curves for passenger cars (U.OF I.Exp. Sta. Bulletin 110), but the variation with weight is not correspondingly great. The formula of Mr. Cole (Marks p. 1225) is satisfactory for use with passenger trains. It will be objected that both of these fit American railway conditions, and this is of course true. However, the largest variation between American and Chinese conditions probably is in the solidity of the track, affectingrolling friction; this is offset by the use of larger car wheels in China, reducing the deflections.

LI. STARTING RESISTANCE. The starting resistance is a very difficult quantity to measure by any but the most refined methods, on account of the acceleration occurring simultaneously. Another difficulty is that all of the wheels must start to rotate at the same time—hence there must be no lost motion or play within the truck or in the centerpins of the car, and if more than one car is used for the test the coupler springs must be blocked and the coupler faces wedged so that slipping or sliding is impossible. Otherwise while starting resistance is being measured on one axle, some ether axles may not be in motion at all yet others be actually rolling. The accepted value for pure starting resistance is 18 lb. per short ton. If there is slack in the couplers the average resistance of the train in starting will be much lower than this figure: at the lowest speeds inherent resistsnces may be roughly represented by the figure of 6 lb. per ton, and for a train getting into motion the starting resistance of the entire train may be safely taken as an average between the two, that is, 12 lb. per ton. It should be noted that it is fully expected to start heavy trains in this way, but the comfort of the travellers demands that locomotives should be strong enough to start passenger trains under the most difficult conditions without jerking out the slack.

LII. INCIDENTAL RESISTANCES. ACCELERATION. This is the most important incidental resistance under ordinary operating conditions. Unlike the elements of inherent resistance, it may be accurately evaluated. For translatory motion we know that $F = Ma$, and this may be reduced to railway units by substituting the mass of one ton and acceleration in terms of miles per hour per seonnd, hence $F=91$ lb. per ton. The figure of 91 lb. per ton for one mile per hour per second acceleration accounts for translational acceleration only, but the rotational acceleration of the wheels must also be provided for. For angular acceleration, $M=J\ w$, where w is the acceleration in radians per second per second, and J the polar inertia. Also angular acceleration may be expressed in terms of radius and linear acceleration : $w = 1.466\ A/r$. The average weight for a pair of car-wheels and their axle is 2200 lb. and the average radius of gyration .64r, hence

$$J = M\,k^2 = 2200 \times (.64r)^2/32 = 28.2r^2$$

The moment to produce an acceleration is therefor

$$M = Fr = Jw = \frac{28.2r^2 \times 1.466A}{r}, = 41\ A \text{ lb. per axle}$$

The complete expression for acceleration is therefor

Total $F = (\text{Resistance}) = 91\ AW + 41\ AN$ ($N$ = no. axles in train)

$$R \text{ (lb. per ton)} = 91\ A + 41\ AN/W$$

Taking $W/N$ as 5 tons, a typical figure for China,

$$R = (91 + 8.2)\ A \text{ lb. per ton.}$$

For the locomotive alone on account of itslarger wheels the figure is higher, namely 110 lb. or slightly more. For an average combination of engine and train the figure 100 lb. per ton serves well.

LIII. OTHER INCIDENTAL RESISTANCES. Grade resistance is merely the weight of the train or car multiplied by the tangent of the angle of rise. Grades are specified as a rise of "N feet per hundred" or "One foot in N feet" or "N meters in 1000" —all of these figures being readily interconvertible. When grade is specfied in "percent" (American practce) the train resistance on the grade is 20 lb. per short ton for each percent. of grade.

The resistance due to curvature is a very uncertain quantity and depends on many obscure variables. Average figures for a locomotive of wheel base of moderate length is about .8 lb. per ton per degree of curvature : for eight-wheeled cars, .5 lb. per ton. The principal variables are speed, weight, rigid wheel base, widening of the gauge, and super-elevation ; no successful formulation or generallization has yet been made.

The additional resistance due to low temperature is not susceptible to formulation. The additional tractive effort required varies so with local conditions of temperature, rail, moisture, etc. that only long experience in a given locality is an accurate guide. It is customary to make "W & W" alloawances (wind and weather) from 10 to 40% in preliminary design calculation.

## SPECIAL LOCOMOTIVES,

LIV. SPECIAL CONDITIONS. While the steam locomotive of the conventional type when well designed and constructed is an engineering achievement of high order, no machine with so unsatisfactory a degree of thermal efficiency could (or should) escape constant critism and efforts at betterment. These efforts at imporivement may be divided into three classes : the attempt to improve conditions as a whole by the substitution of a radically different device ; efforts to improve the conventional style of locomotive by modifications of the thermal principles of operation and the same with the mechanical principles.

In the first class of course the electric locmotive must be considered. From a thermal standpoint the electric locomotive is substantially twice as efficient as the steam locomotive. There are many other elements which must be considered before the "commercial efficiency" can be determined :

Each electric locomotive costs nearly double the price of a steam locomotive of the same power, but fewer units are required.

The repairs on each locomotive are much larger than on the steam locomotive, but there is not a great difference in repairs per mile.

The ultimate question in the case resolves itself into the following problem : Will the coal bill reduction pay interest and amortization on the large outlay for electrification?

Fair—mindedness should be a fundamental characteristic of every engineer : he should be open to appreciate the advantage of anything new. But he should carefully analyze the meaning of tests and not "junp at conclusions" that may be absolutely unfounded. As a case in point the famous" bucking test" in which an electric locomotive out—pulls two steam locomotives is suggested as one which proves nothing and leads to false conclusions on the part of one who is not careful in his analyses.

Every steam engineer recognizes that the electric locomotive has very definite spheres of usefulness :

1. Where the cost of coal is excessive, so that the saving in fuel is a predominant item in any analysis.

2. Where the cost of electric power in very low as a result of available water power.

3. In large cities where the smoke nuisance and noise of the steam locomotive are undesirable.

4. In congested traffic districts, where electrification increases track capacity.

5. On grades so severe that the handling of an adequate train at reasonable speed is beyond the capacity of the steam locomotive.

Electric locomotives have been built under various electrification systems, the most important being A. C. single and three phase, and high voltage D. C. The latter, utilizing voltages of around 3, 000 is the most used at present. In general, the study of the electric locomotive may not be considered as within the scope of this course.

The only other radical substitution which has been made or attempted is the use of the Diesel engine. This of course gives the locomotive that complete independence that is so desirable, but thus far the experiments are only partially successful. The weight of the machine has not yet been brought down to a figure comparable with that of the ordinary steam locomotive for an equal amount of power, and the tendency of any gas engine to "stall" is a strong argument against it.

LV. MODIFIED STEAM LOCOMOTIVES. The basic principles of the first successful steam locomotive were the multitubular boiler, induced draft produced by means of the exhaust jet, two cylinders working cranks 90 degrees apart, and a combined variable cutoff and reversing apparatus. For nearly half a century there was no generally accepted change in these principles, but since 1880 many modifications have been experimented with, only one of which has found general approval and adoption. The modifications may be classified under two heads, thermal and mechanical, though most of them involve both kinds of changes. The thermal modifications are as follows:

The use of superheated steam, which has found general favor and is used on all new engines at present except the very smallest.

The use of the feed-water heater: the heater has demonstrated its uefulness and is making slow headway: it will doubtless become a standard feature in the next few years.

The use of compound cylinders. There was much experimenting with compound cylinders from 1880 until about 1908, after which the greater utility of the superheater in making practically the same saving lessened the interest. Recently there has been a renewal of interest in the effort to find means of further improvement of the economy. Various arrangements of compound cylinders were used, locomotives being built with 2, 3, 4, and 6 cylinders, the four cylinder types being the most successful.

The turbine locomotive at present is being thoroughly investigated. As yet, the same remarks apply as were made in the case of the Diesel locomotive: too much weight for the power obtained. The starting torque problem is another very serious one, as well as the provision of a satisfactory reducing-gear within the available space.

The mechanical modifications: all of the compound locomotives involve mechanical modifications as well, though these are rather matters of detail than principle. The four cylinder and three cylinder "simples" are in the same class. The most important modification in principle is found in the geared types of locomotives, of which the predominant example is the Shay engine. (As used on Nankow pass.) The Shay engine has three vertical cylinders which drive a flexible cranked shaft, which extends the length of the engine. The engine has two, three or four-wheeled trucks, and on the face of each wheel on the right side of the engine is a bevel gear which meshes with a pinion on the crank-shaft. The tractive effort of this engine is therefor

$$To = \frac{1.5\ p\ l\ d^2}{D} \times G \ .$$

where G is the gear ratio (usually 2 or 2.5 to 1) and the other letters have their usual meanings.

Other forms of geared locomotives are in use: the rack locomotive and the combined rack-and-adhesion engine, used for service on grades too steep for the adhesion engine to be successful. The rack type engines have the drive through a gear or set of gears placed between the wheels, and this gear meshes with the rack which is part of the track. Such locomotives are usually built with the center line of the boiler on a slant, in order that it may be approximately level when the engine is running upon the average grade for which it is designed. Ordinary adhesion engines are built in small sizes with gear drives, both with inclined cylinders and a bevel gear and pinion arrangement, and with the standard arrangement of cylinders and rods, except that the connecting rod drives the crank on a pinion shaft, the pinion meshing with a gear on the main axle. It is difficult to see any compensation for the complication thus introduced.

The mechanical modifications due to compounding must be noticed. Compounds may be classified by the number of cylinders. They are in most cases called by the names of the inventors.

1. *Two Cylinders*: High pressure on one side, (usually the left) and low pressure on the other side, in the normal positions. The high pressure cylinder exhausts into a large receiver which serves as the connection between the two. This is the type commonly called the "cross-compound".

2. *Three Cylinders*. a. Three cylinders of the same size, the high pressure between the frames, and the two low pressure cylinders outside of the frames. (b.) Two high pressure cylinders outside the frames and the two exhausting into a large low pressure cylinder between the frames. These are known as the French and English types of compounds respectively.

3. *Four Cylinders.* This is the largest class and needs to be further subdivided into balanced and non balanced types. In the balanced types the two adjacent cylinders, one high and one low-pressure work cranks 180 degrees apart, thus very nearly balancing the inertia forces and producing a very smooth-running engine at high speeds. The principal types of balanced compounds are: the von Borries, having four cylinders in the same line, the H.P. outside, the L.P. between the frames, and the Baldwin, with the arrangement reversed; (all cylinders drive the same axle in these cases); the de Glehn, having the low pressure cylinders inside and under the smoke box, driving the front axle and the high pressure cylinders outside and several feet further back, driving the second axle; the Cole type with the low pressure cylinders in the normal position, and the high pressure cylinders above the front bumper bracket or foot-plate, also a "divided drive" engine. It is interesting to note that this is the cylinder arrangement used for the majority of the British "four cylinder simples". The non-balanced types are as follows: the various forms of articulated engines, described in more detail in a later paragraph; the tandem type, where there are a highpressure and a low pressure piston on the same rod on each side of the engine; the Vauclain type, where there are separate pistons and rods but two cylinders, one high and one low, are coupled to the same crosshead.

4. *Six Cylinders.* Occurs only in the case of the "triplex" articulated engine.

LVI. ARTICULATED TYPES. The articulated type of locomotive has been developed to give flexibility on sharp curves, and in general consists of two (three in the case of the Triplex type) more or less independent sets of running gear supplied with steam by a single boiler. Minor types such as the Farlie, Meyer, Fechot, and Garratt, may be passed over, and the Mallet type may be studied with attention to detail. The general plan of the engine is shown in Fig. 9A. The boiler is of large size, and under the back end the rear or high pressure unit is located, and is fastened rigidly to the boiler. The front or low pressure engine is connected to the rear unit only by the hinge pin H, and through the ball-jointed steam pipe R. The front end of the boiler rests on two or more bearing plates, on which it can slide laterally when the engine rounsd a curve. One of the bearing plates is fitted with springs which exert a lateral pressure tending to restore the boiler and running gear to their normal relation. Notice the steam passages : when the engine is running under normal conditions the steam passes into the dry pipe, through the superheater, and through the main steam pipes L—L back to the high pressure cylinders. After expansion here it is exhausted into the receiver R—R, which has a flexible connection to the high prssure saddle. Steam passes from the receiver through the Y—connection Y into the low pressure cylinders, whence it is exhausted into the smoke box through the exhaust pipe X, which has flexible connections at each end. Notice that only low pressure and exhaust steam passes though the flexible jointe. When the engine starts it is necessary to turn high pressure steam through a pressure-reducing valve into the low pressure cylinder, and this is managed by the use of an "intercepting valve" placed in the high pressure saddle. This steam may be exhausted into the receiver, or direct to the atmosphere according to the position of the valve. The pipe AA provides the direct path to the atmosphere. In the Mallet type the steam locomotive may be considered to have reached its highest development from the standpoint of a power-plant ; space is ample for economizing devices. and the boiler proportions are such that efficiency should be high. However, much of the thermal advantage is lost as a result of excessive internal friction, and the only field in which the Mallet is an economic factor is slow and heavy goods service.

### LVII. MECHANICS OF THE COMPOUND ENGINE.

There are various methods of figuring the tractive effort of the compound engine, and so much depends on operating skill of the crew that any figures are more or less uncertain. The usual basis is the assumption that the work is to be evenly divided between the high pressure and low pressure cylinders. Steam is admitted into the high pressure cylinder at approximately boiler pressure $P$, and exhausts into the low pressure cylinders at pressure $p$.

Ignoring the wire-drawing losses, and letting dh represent the diameter of the high pressure cylinder and dl of the low pressure cylinder.

$$(P-p)dh^2 = pdl^2, \text{ hence} \frac{P-p}{P} (dl/dh)^2$$

The quantity $(dl/dh)^2$ is the ratio of the areas of the high and low pressure cylinders, and is commonly called R. Hence,

$$\frac{P-p}{p} = \frac{P}{p} - 1 = R, \text{ hence } p = P/(R-1)$$

Substituting the figure for the available pressure just obtained in the usual tractive effort formula, the tractive effort for the engine in terms of the diameter of the low pressure cylinder is

$$To = P - \frac{dl^2 l}{(R-1)D}$$

For a four cylinder engine this must be doubled. The tractive effort working simple is dependent on several factors of design and especially the arrangement of the intercepting valve. In general, the valve which admits steam directly from the boiler to the low pressure cylinder reduces the pressure to such an extent that the low pressure piston thrust is equal to the high pressure thrust. In this case the tractive effort will be

$$Ts = \frac{P\ ph^2 l}{D}$$

and the ratio of the two tractive efforts is

$$\frac{Ts}{To} = (dh/dl)^2\ (R-1) = \frac{R-1}{R}$$

Since the usual ratios between the cylinder sizes are from 2.25 to 2.5 this would indicate a gain of approximately 40% due to working simple. This full amount is however not realized; the back pressure in the High pressure cylinder is excessive due to the small size of the direct exhaust line; sometimes a small amount of high pressure steam is admiited behind the high pressure piston to protect the parts from overloading; the receiver relief valves are set lower than the cylinder ratio would dictate. The usual allowance for working simple is 20 or 25%.

# Notes on Railway Mechanical Engineeing

## PART II.

### THE SPRING RIGGING AND WEIGHT DISTRIBUTION.

1. THREE–POINT SUPPORT. The simplest form of wheeled vehicle (where the load is entirely carried by the wheels) is the three-wheeled cart. Each wheel carries a portion of the load at all times, depending on the shape of the wheel-base, and the amount and arrangement of the load. If four independent wheels are used, one of the four may carry no load, since four points are not in general co-planar. However, if two of the wheels are mounted on an axle and the load from the body of the vehicle is transmitted to the center of this axle, a condition of three-point support is restored, and all of the wheels will carry a share of the load. By various extensions of the above scheme practically any wheel arrangement may be resolved into a three-point support system, and such an arrangement is regarded as essential by American locomotive builders. Partial "equalization" is practiced by the Europen builders. An interesting illustration of the principle is seen in the methods of carrier coolies in handling a heavy load : notice the manner in which the bamboo poles serve as "equalizers". When springs are interposed between the wheel and the load, the wheel can not entirely escape its share unless the depression in the track is sufficient to allow the spring to extend to its "free height"—nevertheless equalization is an advantage even on the best track.

II. VIRTUAL AND ACTUAL SUPPORTS. Consider the case of a pair of locomotive driving wheels as in Fig. 10, equalized by the equalizer beam acting on the ends of the springs. Neglecting the "dead" weight of the wheels, axles and journals, the weight of the load imposed at the actual points of support A, B and C is the same as the weight on the wheels at D and E, and if the equalizer has equal arms, the center of gravity of the entire system—either of the actual supports or the of the virtual supports—that is, the wheels, is at B. Now suppose that it is desired to have the weight on the left wheel double that on the right wheel. The equalizer arm is now divided in the ratio 2 : 1, B the fulcrum being moved toward wheel D. Since the springs are symmetrical the loads on the hangers at the ends of each spring are the same, and the two inner hangers have their center of support at B it is obvious that *the center of the actual support is* still *identical with that of the equivalent supports,* though it is no longer located at B.

III. THREE—POINT SUPPORT AS APPLIED TO LOCOMOTIVE. In a fully equalized locomotive the arrangement of the supports is usually as follows :

Six-driver type : front pair of drivers cross—equalized to form the front supports ; second and third pairs equalized together on each side, but not cross equalized.

Eight—wheel or Ten—wheel types : front support formed by the center pin of the engine truck, and drivers on each side equalized together, but not cross equalized.

Atlantic or Pacific types : same as foregoing, with trailer equalized with driving wheels. Where the trailer is used it is usually necessary to use a cross beam back of the rear drivers, but the connections are so arranged that there is no cross equalization.

Mogul type : engine truck and front pair of drivers equalized together and cross equalized. Back two pairs of drivers on each side equalized but not cross equalized.

Consolidation type : engine truck and front two pairs equalized and cross-equalized, rear two pairs as before.

Prairie and Mikado types : same as foregoing except trailing wheel equalized with rear driving-wheel system.

Draw diagrams in perspective to illustrate each of these.

IV. SPRING ARRANGEMENT. There are two arrangements of the spring rigging in common use. When the plate frame is used it is necessary to place the spring *Under* the journal box, the weight being transmitted through a link connected to the lower

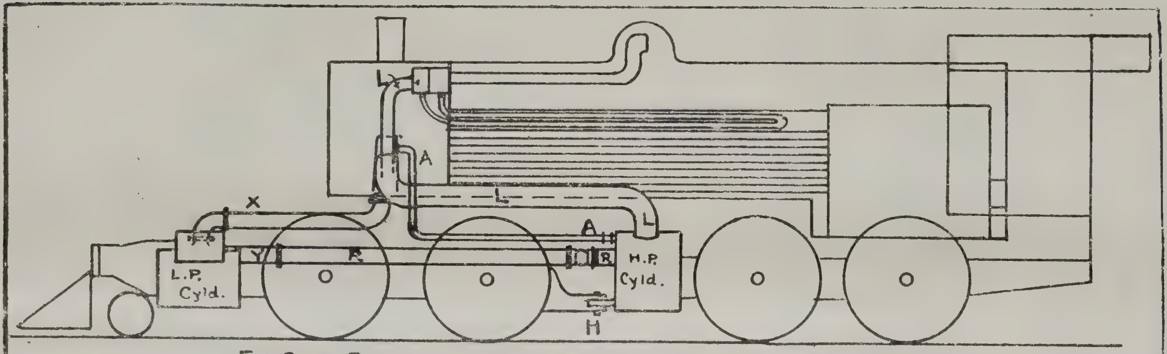


FIG. 9A ~ DIAGRAM OF MALLET ARTICULATED LOCOMOTIVE.

FIG 10
VIRTUAL AND ACTUAL POINTS OF SUPPORT

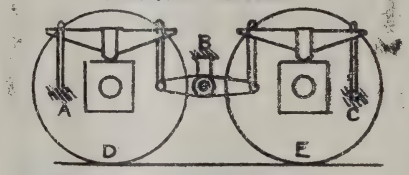
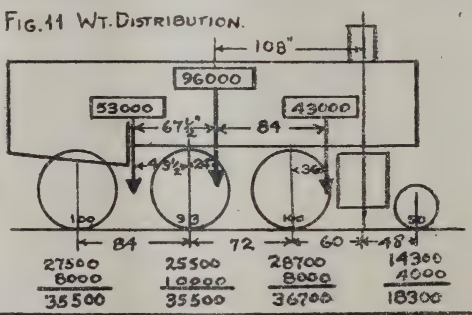


FIG. 11 WT. DISTRIBUTION.



part of the box. The bar frame is usually accompanied by the *overhung* type of spring—that is the spring (or the equalizer which is substituted for it where room for a spring is lacking) rests on top of the journal box. Fig. 1 of Part 1 shows an underhung spring rigging, and Fig. 10 of Part II shows the overhung type, as does also Eig. 12.

V. WEIGHT AND CENTER OF GRAVITY OF THE ENGINE. As a part of the design work on every locomotive a weight estimate must be made, and the details and process for this will be studied at length later. Along with the weight, the center of gravity of each part is noted and finally the center of gravity of the entire locomotive (omitting the dead weight) is found. The spring weight thus found is finally distributed to the various supporting centers and finally to the wheels, where by the addition of the dead weight the weight on the rail is found. To illustrate the process in its entirety a complete example will be given.

VI. ILLUSTRATIVE EXAMPLE. In Fig. 11 is shown the outline of a mogul locomotive. It is estimated that this locomotive will weigh 126,000 lb., of which 30,000 lb. is dead weight and 96,000 lb. is carried on the springs. It is desired to have the driving wheel weights equal and approximately 35,000 lb. each. The center of gravity of the spring weight is 108" back of the saddle center. From the relation which will exist between the final rail weights it is evident that the spring weight upon the engine truck will be about half that on the front driving wheel, and it may therefor be labled (50%). In order that the rail weights on the back pairs of drivers may be equal it is evident that the spring weight on the middle pair must be 2000 lb. lower than that on the back pair—this 2000 lb. will represent about a 7% reduction in the spring weight, hence the middle wheels are labeled (93%). Note that the engine truck and front drivers form one support, while the two rear pairs of wheels form two supports, one on each side of the engine, and of course superposed in the sketch.

## VII. DISTRIBUTION OF THE WEIGHT.

1. Virtual center of front support : $\left(\frac{50 \times 108}{150}\right) = 36''$ ahead of front driving axle.

2. Virtual center of rear supports : $\frac{100 \times 84}{193} = 43.5''$ back of middle axle.

3. Distance from center of gravity of spring weight to front support 84″, to back support 67.5″, hence division of weight between the two : on front support, $\frac{96{,}000 \times 67.5}{67.5 + 84} = 42{,}800$ lb.

on back support, $\frac{96{,}000 \times 84}{67.5 + 84} = 53{,}200$ lb.

4. Distribuiton to axles :

front support, front truck, 42, 800 × 50/150 = 14, 300 lb
driving axle, 800 × 100/150 = 28, 700 lb.
back support, middle axle, 53, 200 × 93/193 = 25,600 lb.
back axle, 53, 200 × 100/193 = 27,600 lb.

Note that these distributed figures should add up to the total spring weight. No calculations need be carried closer than the nearest hundred pounds.

5. Finally, add dead and spring weights together.

It will be noticed that the driver weights in the two systems vary about a thousand pounds. There are two possible methods for correcting this. There are usually some parts on the engine which can be altered in position to some extent, and usually the castings at the extreme ends may be slightly varied in weight. If the center of gravity of the engine could be moved back an inch further the axle distribution will be practically equal. Another method is that of modifying the supporting system : an increase of the percentage load on the engine truck will have a *double* effect in reducing the spring weight falling on the driver. Try increasing the percentage from 50 to 55.

VIII. THE SPRING RIGGING. The spring rigging arrangement for this engine is shown in detail in Fig. 12. The

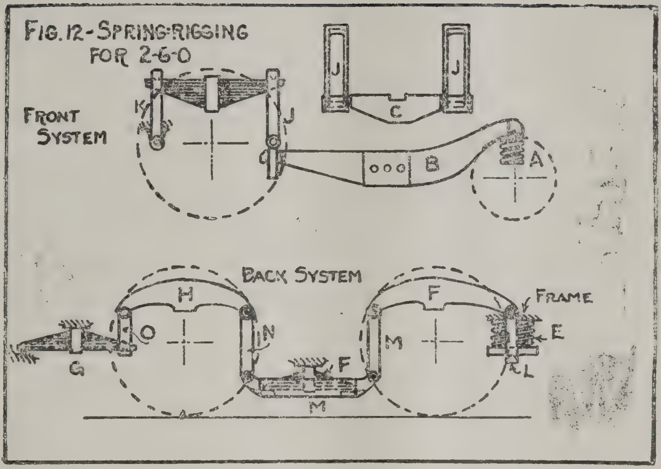


FIG. 12 - SPRING-RIGGING FOR 2-6-0

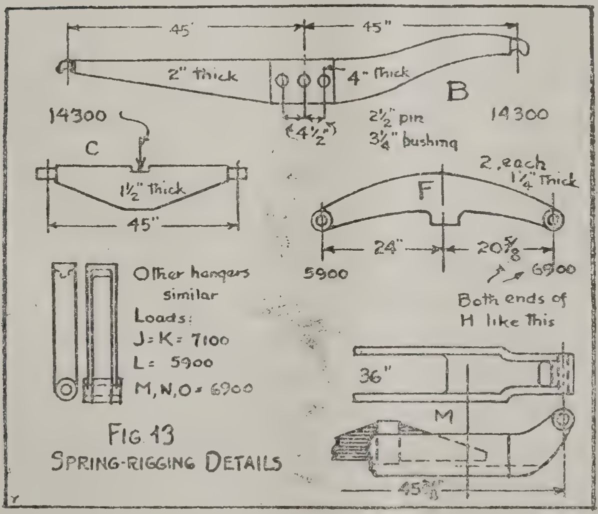


FIG. 13
SPRING-RIGGING DETAILS

engine truck has two coil springs A. The long equalizer B is connected to a cross equalizer C which in turn is connected to the two springs D over the front driver. There is not room over the middle and back drivers to place springs, hence the arrangement of "between-frames" springs must be used. In the case of the front support only the position of the fulcrum needs to be determined. The engine truck equalizer being 90″ long and the load on hanger J as well as on the truck being 50% the fulcum point is obviously at the center of its length, 3″ ahead of the cylinder saddle center. The design of the rear section is much more difficult. The designer is confronted with several limitations: the longest spring which can be used between the frames (and give suitable deflection) is 30″. The minimum overhang of the equalizer over the middle driver is 24″ ahead of the wheel, and a longer overhang than this is undesirable on account of the difficulty in obtaining sufficient strength. Since the load on the middle driver is to be 93%, call the load on hanger L 43%, and make the arm carrying hanger L 24″ long. Then if the other end of F is made 20 5/8″ long, the load on hanger M will be 50%, and the combination gives the desired 93%, Now the equalizer H may be made equal armed, of say 36″ length, and the connection between M and N may be fulcrumed in the center. This connection mustbe 84—20 5/8—18″ = 45 3/8″ long. A plain equalizer could be used but the engine would probably ride very hard with this arrangement. A better plan is to connect the two with a "cradle" equalizer into which a plate spring may be set. In the case of a mogul there would not normally be room for such a device between the upper and lower frame rails (see Dictionary for the reason), so the hangers may be carried down below the lower frame rail.

IX. THE SPRINGS. The springs as used on this engine are as follows: Engine truck springs A, two coil springs carrying a load of 14,300 lb., following the distribution as made. This gives 7,150 lb. per spring, and about 5% additional should be allowed—say 7,500 lb. Front driver spring D, two springs carrying 28,700 lb.; with 5% allowance the load on each spring is 15,000 lb. Coil spring E—one or more coil springs on each side, carrying 43%

of the unit load on the back section, or 43 % × 27,600 lb. = 11,850 lb. or say 12,500 lb. with 5 % allowance on each set of springs. Plate springs G and F, two springs on each side, each pair of springs carrying the unit load of 27,600 lb. hence one spring with its allowance should be designed for 14,500 lb.

X. DESIGN OF COIL SPRINGS. The following formulae are useful in the design of coil springs:

The load when the spring is compressed = $S\pi d^3/16R$, where d is the diameter of the steel stock and R the mean radius of the coil. Locomotive and other railway springs are usually designed to "go solid" at about 80,000 lb. strees, an amount considerably below the elastic limit of the material. Hence the load is practically 16,000 $d^3/R$ when the spring is solid. The load on the spring should be about half of this figure—that is, a working stress of 40,000 lb. (Verify.)

The "free" height of the spring per inch of SOLID Height = .08 $R^2/d^2$ for it to go solid under a load producing 80000 lb. stress. Actually, spring design work is generally done from tables based on these or equivalent formulae. Such tables are available in most handbooks and drafting rooms.

XI. DESIGN OF PLATE SPRINGS. The load per inch of width of a single plate, with the stress 80,000 lb., is 53333 $h^2/L$ where L is the length between hangers, and h the thickness of the plate.

The deflection for a spring of the usual construction, that is, with 25% of the plates full length, and the shortest plate about one fourth of the nominal length is .00061 $L^2/h$. Spring tables for plate springs are usually available without the necessity of making the calculations.

XII. LIMITATIONS OF SPRING DESIGN. These are of two types: those resulting from the available space and those from the nature of the springs. In the present example, the engine truck spring on a two-spring truck can not usually be made more than 8″ in diameter at the outside, and a smaller diameter is preferable. Coil springs can always be nested—that is, one spring be placed inside of the other, but it is essential that the deflection *rates* of the two springs be equal or the load will not divide as planned and one spring will be overloaded. Similar remarks apply to the spring E, except that the outside diameter should not be greater than about six inches, and less if more than one coil is used, as they must be placed one ahead of the other.

The only limitation on the plate springs is that of width in the case of spring F. By placing the cradle below the lower frame rail it may be made about six inches wide ; allowing one inch for the sides, and 1/2″ for the spring-band the plates themselves may be 3 1/2″ in width.

*The Deflection of all of the Springs on the Engine Should be Approximately Equal Under Normal Working Load.*

### XIII. DESIGN OF SPRINGS For EXAMPLE.

| Spring | Load | Length | Thickness | No. Plates | Width | Cap'y | Defl. |
|---|---|---|---|---|---|---|---|
| D | 15,000 | 36″ | 7/16″ | 13 | 4 | 14,950 | 1.81 |
| F | 14,500 | 30″ | 5/16″ | 24 | 3 1/2″ | 14,500 | 1.76 |
| G | 14,500 | 30″ | 5/16″ | 19 | 4 1/2″ | 14,750 | 1.76 |
| A | 7,500 | lb. | | | | | |

Use an outside coil 7″ outside diamter, 1 1/4″ steel ; load "solid" 10,700 lb. Deflection ratio, 1.42 ; inside coil 1 1/4″ O.D., 13/16″ steel, load solid 4,900 lb., deflection ratio, 1.36 ; average deflection ratio 1.39, total load solid, 15,600 lb., load half solid, 7,800 lb.

E 12,500 lb. Use two springs, each 6,250 lb. Outside coil, 6″, 1 1/8″ steel, deflection ratio 1.37, load solid, 9,900 lb. ; inside coil, 3 1/2″ O.D., 11/16″ steel, load solid 3600 lb. Total load solid 12,800 lb., half solid 6,400 Average deflection ratio 1.35.

Now the total deflection of the coil springs must be equal to that of the plate springs in order for the engine to ride level and the load to distribute properly. This average deflection may be taken as 1.8″. The coil springs have a deflection ratio of 1.39 for the engine truck; that is, the free spring is 1.39 times as high as the solid spring, hence the half solid spring is 1.185/1.39 × free height and the deflection under the half-solid-load is .15″ per inch of free height. This requires a total free height of 12″, or a half solid height of 10″. The figure for spring E is practically the same.

XIV. DESIGN OF THE EQUALIZERS, HANGERS AND PINS. Spring Rigging pins are designed for about 3,000 lb. bearing pressure and 4,000 lb. shear. The holes in the equalizers are usually bushed with steel tubing. The engine truck equalizer is usually provided with two or three pin holes to permit alteration of the weight distribution. Hangers are safe with 5,000 to 6,000 lb. tensile stress. Forged equalizers are stressed to 12,000 lb and cast steel beams to 6,500 lb. It is to be noted that there is usually a boss as the center of an equalizer, and the maximum stress rarely occurs at the fulcrum for this reason—the weak point is usually at the end of the boss where the width is reduced, and generous fillets should be used at this point.

In Fig. 13 the details of the spring rigging is shown complete with the loads carried by each part.

## RAILWAY BRAKES.

XV. METHODS OF RETARDING THE MOTION OF WHEELED VEHICLES. There are several different possible methods by which the speed of a moving vehicle may be decreased: "drags", "spragging", "the kickback", etc., but only one, namely the method of applying a force to the rim of the wheel or a concentric drum and dissipating the energy of the vehicle in heat, is practical for general use. The actual retarding force is the pressure of the block or shoe on the rim multiplied by the coefficient of friction—this force acts precisely as if it were applied at the drawbar to change the speed. (Why? A certain amount of energy in the moving car must be overcome by the application of a certain amount of work. The travel of a point on the rim relative to the brake shoe on the car is equal to the travel of the car relative to the track, providing of course that the wheels do not slip, for which condition see further discussion. Hence to overcome the same amount of energy, that is, to stop the car under the same conditions since the work is equal and the distance equal, the force is equal. This explains why a brake shoe placed at any point of the rim is equally effective, except insofar as the weight is changed.

Knowing the braking force and the coefficient of friction, the retarding force, which is the product of the two, acts according to all of the laws of traction, as developed in a previous section. For example, if the retarding force on a car is 90 lb. per ton, and the *down grade* one percent., (20 lb. per ton) and the inherent resistance 8 lb. per ton, the resultant net force is $90—20+8=78$ lb. per ton, which would give a retardation of .78 m.p.h.p.s. Sometimes the effect of inherent resistance is ignored in order to be conservative and safe in the calculation of retardation.

XVI. EFFECT ON WEIGHT. In Fig. 1, part I, note the brake hanger and shoe arrangement. When the engine is running forward and the brakes are applied there will be a compression in the hanger, hence a lifting effect on the portion of the locomotive above the springs: that is,a greater proportion of the weight becomes "dead weight" since a portion(=the braking force if the brake hangers are vertical) of the weight which is above the springs is carried to the track through the brake hangers. The reverse of this effect is very important, because the pulling down may overload the springs. There is however an other effect on weight to be noted. Fig. 14 shows the outline of a 4—4—0 engine, with weights and distribution as indicated. There is a difference of 48″ in the height of the center of gravity of the spring-borne parts and the point of application of the retarding force, hence there is a couple tending to revolve the spring-borne parts clockwise of 6,000 lb. (the total retarding force) × 48″. This couple is counterbalanced by an increased pressure on the engine truck, and since the mean distance of the truck center from the center of gravity of the engine is 142″ the additional pressure on the truck is 2000 lb., and the rail load on the driving wheels is therefor reduced by this amount. There is also a redistribution of the spring weights : the new center of gravity is $3^1/_2$″ ahead, but the total rail weights are not affected by this.

XVII. SLIPPING. It is necessary for rim or drum braking that there must be no sliding between the rim and the track. The static coefficient of friction between two steel surfaces ranges from 18 to 30%, while the sliding friction coefficient is about 15% for ordinarily smooth surfaces. Now if the product of the braking force and the coefficient between shoe and rim is greater than the product of weight by the static coefficient, the wheel will cease to slide past the shoe and begin to slide on the rail. The latter is very undesirable for two reasons : the sliding of the rim on the tire quickly results in a flat spot, which requires turning off, and the effective retarding force is greatly reduced. In designing brakes therefor the weight must be taken into careful consideration and the possible variations due to the application of the braking force, as developed in the preceding paragraph. Note that the use of two shoes on each wheel practically eliminated this particular difficulty but the practice is not often resorted to on locomotives. The usual design figures for braking force are as follows :

For locomotives, 60% of the weight on the braked wheel in working order.

For tenders, 100% of the empty or "light" weight.

For passenger cars, 75% of the light weight.

For freight cars, 60% of the light weight.

XVIII. THE COEFFICIENT OF FRICTION. The coefficient of friction between the shoe and the rim varies with several different criteria : the shoe pressure, the speed, the length of time of application, and the material of shoe and rim. The latter may be dismissed for general purposes, especially in China,

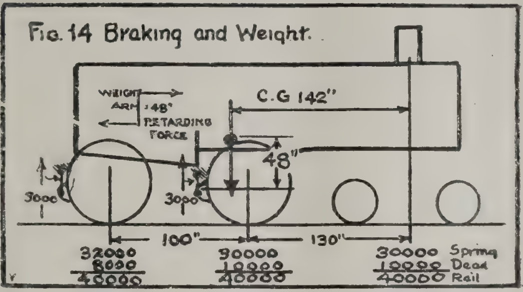


Fig. 14 Braking and Weight.

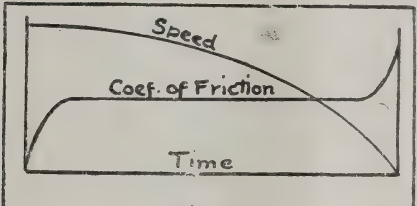


Fig. 15

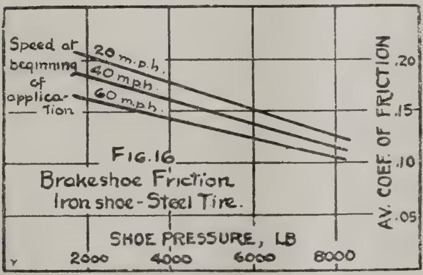
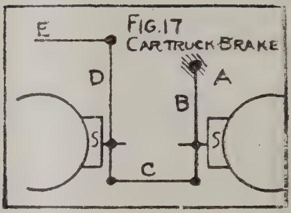


Fig. 16
Brakeshoe Friction
Iron shoe - Steel Tire.

FIG. 17 CAR TRUCK BRAKE



where even on goods cars the steel tire is universal. Shoes other than plain cast iron are very rarely used at present though there has been much experimenting with special materials, abrasive inserts, etc. The variation with application time is as follows: there is a sharp decrease in the coeffient for a few seconds after the application, after which it continues to decrease slowly, if the speed remains constant. If the speed decreases, other things being equal the coefficient increases, until just before rotation ceases there is a sudden large increase. It is thus seen that for constant pressure, the lengthening time of application and the decreasing speed tend to counteract each other to keep the coefficient constant; however, the speed effect is slightly greater, and the coefficient increases slightly. The characteristic coefficient curve for an average stop with constant pressure on the shoe is shown in Fig. 15 which also shows the speed relation. Fig. 16 shows the former standard specifcations of the M. C. B. Association for cast iron shoes with varying pressures. For conditions in China the following representative coefficients may be used: for locomotives, .12, for freight cars, .20, for passenger cars, .15, all based on slowing down from speeds of 20 or 30 miles per hour.

XIX. HAND BRAKES. The original method of applying the shoes to the rims with the requisite pressure was a plain lever, and this arrangement is still seen: often so arranged that a man may stand on the handle. At present most car brakes are applied by some variation of the "floating lever" system as shown in Fig. 17. "A" is a fixed fulcrum on the truck frame; the shoes S respresent two shoes each on opposite wheels, the vertical levers actually being connected to the brake beam between them. The lever D is the floating lever, having no fixed point. The force is delivered to the system through the pull rod: this may be produced by a chain wrapping on a drum, by a lever, by a crosshead running on a screw, or otherwise.

XX. POWER BRAKES. There are three media which are or have been used to operate the brake system. These are steam, vacuum, and compressed air. Steam for obvious reasons is suitable only for the brakes of the locomotive itself. The vacuum system is used on most British and some other railways, elsewhere the airpressure system. The two have many points of analogy. The pressure system has a great advantage when applied to heavy equipment, on account of the smaller sizes of the operating cylinders. There are two types of air pressure equipment: the "straight air" and the "automatic air". The former is the original. In this system the air-pump compresses the air into the main reservoir; to apply the brakes a valve is opened, and air pressure is conducted through the brake pipe to the cylinders on the various cars. This type of brake is only suitable for trains of two or three cars at the most for the following reasons:

The brakes on the front car are applied first, and the rear cars therefor run up on the forward cars, producing large stresses in the draft gears.

The entire brake system becomes inoperative leaving the train unprotected in case of a break in the line. The break will go undiscovered until the engineman attempts to make an application.

The brakes can only be applied by the engineman.
The second type, which is in universal use on steam lines today is the automatic air. In this type, the air for one application is stored in an auxiliary reservoir on each car: the application is made by decreasing the air pressure in the train-line, which decrease actuates a balanced valve, causing the air in the reservoir to flow into the cylinder. The actuating valve is known as the "triple valve" since it has three functions:

Providing a passage from the cylinder to the atmosphere when the brakes are off.

Providing a passage from the the reservior to the cylinder when the brakes are applied.

Providing a means of recharging the reservoir from the train-pipe while the brakes are released.

There are several types of "triples—"the most important of which are the "plain", "improved", and "quick action". The plain valve, used everywhere but now practically obsolete did nothing more than the things named- The "improved" valve is the type built by the British Westinghouse Co., and is generally used in China. This type performs the functions named, with an important modification : if a large reduction in the train pipe pressure is made, a much larger passage is opened between the reservoir and the cylinder, making a much quicker application. In the "quick action" type, there is an additional valve which in case of an "emergency" application also opens a passage from the train itself into the brake cylinder, increasing the force as well as the quickness of the application. The increased force is about forty percent.

The automatic air-brake obviates all of the disadvantages of the straight air type. Note particularly that in case of a break in the train line the brakes are applied in *emergency,* and the train can not proceed until the break has been repaired, The automatic air can be applied from any point in the train by merely opening "bleeders" in the train line. The automatic air brake controls a train of 150 cars without difficulty ; with such a long train there is of course some running-up effect. but a skillful engineman reduces this to an amount which is not detrimental to the equipment.

XXI. AIR BRAKE DETAILS. The brake system of a locomotive consists of two portions, the operating portion and the foundation portion. The foundation brake consists in turn of the hangers, shoes, pull rods, equalizers, etc., the brake beams, and the main crank fulcrum ; the operating portion consists of the main cranks and all of the other parts concerned in condensing and distributing the air. These are as follows :

The Pump. Standard Westinghouse pumps have 8,9 1/2, or 11$''$ cylinders for air and steam, and are also built cross-compound. They are pure reciprocating mechanisms, taking steam full stroke. The steam distribution is controlled by a differential piston valve in the top head of the steam cylinder, and this valve is actuated in turn by air pressure controlled by a plunger rod, the motion of the latter being controlled by the steam cylinder piston. The various pumps differ only in minor details. The air cylinder has two poppe valves opening inward from the atmosphere, and two opening outward to the main reservoir connection.

**The Reservoir.** Air is carried to the reservoir through several return bent lengths of pipe in order that it may cool after the compression. The reservoir is built of quarter—inch boiler steel, with dished ends and single riveted lap—joints. Reservoir capacity is about one cubic inch for each 5 lb. of engine weight.

**The Brake Cylinder.** Ten inch stroke by even diameters from 8 to 18″. Two cylinders used for eight drivers or less. Both cylinders connected to the brake rigging through cranks, working on the same fulcrum.

**The Engineer's Valve.** All of the piping connecting the various parts of the system is headed up in the cab, and, surmounted by a rotary valve which makes the necessary connections, its principal function being the venting of the train-line to the atmosphere.

**The Pump Governor.** Operation of the air pump is controlled by a governor valve which opens automatically when the pressure in the main reservoir is reduced to a predetermined lower limit.

**The Distributing Valve.** Usually located under the engineman's seat below the cab floor. Consists of the ordinary triple valve mechanism, plus an additional mechanism enabling the engine to be operated on straight air, with some additional modifications. The most complicated part of the entire apparatus.

XXII. FOUNDATION BRAKE RIGGING. Foundation brake riggings are of many different types and very little in the way of general design principles may be given. The following points should be observed :

The brake when released permits the shoe to stand at a distance of about 3/4″ from the tire. The desirable distance for the cylinder piston to travel in making the nominal application is from 5 to 6″, hence the total leverage in the system must be between 6.5 and 8.5. After the nominal application has been made the piston will travel about 30 to 40% further straining the rigging, and pulling up all the slack, and the total travel should not be more than 8″, or the equalization pressure will be too low.

In general, both cranks should be keyed to a shaft running across the engine, so that if one cylinder should be out of order it may be cut out and the other cylinder will give a half application on both sides of the engine.

Pull rods may be slanted about 1 in 4 without material reduction in the effectiveness of the brake.

The efficiency of the various forms of foundation brake riggings runs from 90 to 96%. An efficiency assumption is often used in making the calculations.

The rods, jaws, beams and equalizers may be stressed to 15,000 lb. under *service* application of the brake, but no pin or rod of less than 7/8″ diameter should be used. Pin bearing pressure, 3000 lb.

The lowest part of the brake rigging should clear the level of the top of the rail at least 2″ when the engine is down on the boxes——say 5″ minimum clearance when in working order.

Brake shoes are hung so that the top of the shoe is about on the horizontal center-line of the wheel. They MUST be so carried in the hangers so that the pin is in double shear, and so arranged that the top of the shoe does not fall against the tire when the brakes are off.

Fig. 18 shows a typical design of foundation brake rigging for a six-driver engine. Assume that the weight is 36,000 lb. per pair of drivers, the braking force per wheel is 10,800 lb. If the hangers are of 4 : 3 leverage, the force at the bottom of the hanger is 8100 lb. Equalizer E has equal arms, while F has arms in the ratio 2 : 1, the shorter arm connecting to the two back brake beams. The pull in the rod H is 8100lb., in G,16,200 lb., and in A, 24,300 lb. All of these rods should be provided with turnbuckles. Suppose a 10″ cylinder is used : with equalization pressure of 50 lb. (how large an auxiliary reservoir would this require, with a pressure of 70 lb.?) the piston thrust is 3930 lb. and the ratio of the main crank arms 24,300 : 3930 =6.19, hence the total leverage is 4/3 × 6.19 = 8.24 which is satisfactory. The theoretical travel of the piston to make the application is 3/4″ × 8.24 =6.19″ leaving 1.81″ additional travel to take up the spring in the rigging, which is ample for a well maintained rigging.

## THE TENDER.

### XXIII. THE PROBLEMS INVOLVED IN CAR DESIGN.

In general, the construction and design of railway cars is much further standardized than is that of the locomotive, and from the nature of the case it is evident that the car as a whole is a much simpler matter. The ordinary process of design is simply the combination of a well standardized set of details. The tender of the locomotive may be taken as typical of the entire class and briefly studied from a structural standpoint.

XXIV. THE TANK. The tender exists for the tank, and the most satisfactory tender is the one which provides the most suitable tank—meaning thereby one of the requisite capacity, strength and rigidity—combined with a suitable running gear—meaning one which is strong and safe, economical with regard to materials, repairs and operation. The tank is in general a superficially rectangular box, divided into two compartments, one for coal and one for water. The latter is much the larger. The coal space must be open to the cab, of easy access, and it is very desirable that the sides shall be so arranged that all of the fuel will automatically slide down to the front of the space, thus minimizing the distance which the stoker must walk. The angle of repose of coal is about 45 degrees, which fixes minimum slopes for the sides. The tank is generally built of quarter-inch steel plates, except the "shovel-plate" made thicker to protect it from wear, with single

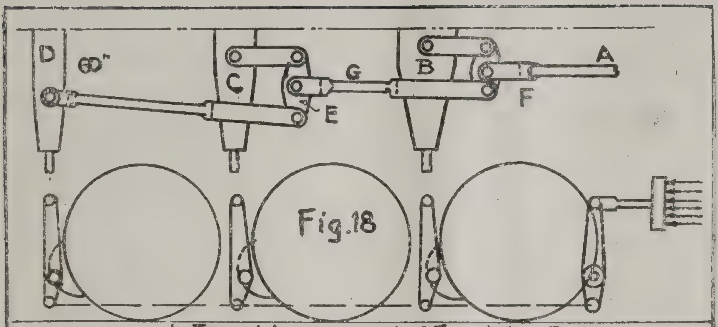


Fig. 18

A Typical Arrangement of Foundation Brake.

Loading of Center-sill.

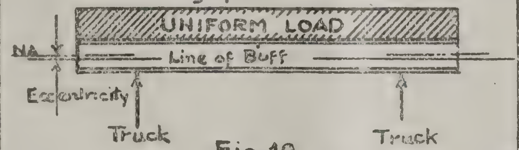


Fig. 19

riveted lap joints. Shovel plate rivets are countersunk. A filling hole is provided at the back on top of the tank; longitudinal and cross plate braces are provided, the latter commonly called "swash plates" since they serve the additional purpose of preventing surges of the water when the speed changes. The sides of the tank are carried up a few inches above the top as a safety device, for it is commonly necessary for the trainmen to climb over the coal, unless the tank is made narrow enough so there is a footing around the edge of the platform,—in this case the rim serves as a hand rail. The sides are usually built up higher at the coal space to increase the capacity. No fixed rules for tank and bunker capacity can be laid down—obviously the capacity required depends on the location of coal and water supplies and the rate of consumption of the engine. In general it may be said that the capacity either of fuel or of water should be sufficient to permit the passing of alternate supply stations, unless the latter are terminals. The ratio between the amounts carried is usually around one and one-half tons of coal for each thousand gallons of water,—that is, 2.5 lb. water per lb. coal.

XXV. THE FRAME. Tender frames usually consist of channels: two at the center, spaced widely enough for the center pin castings and the draw gear to be placed between them (15 to 18″) and two at the outside edge of the frame. These channels are considered as continuous beams with uniformly distributed loads, and supports set a considerable distance in from the ends. The trucks are usually so placed that the weight carried by the two is the same when the tender is about half loaded. Each channel is considered as supporting the weight half—way to the adjacent channel on either side. The center channels in addition must carry an eccentric buffing or jerking force. The former is much the larger, and should be taken at about five times the tractive effort of the engine. Great care must be taken in the analysis that stresses that are cumulative should not be overlooked. Fig. 19 shows a typical loading diagram for the center sill.

The remainder of the frame consists of the transom castings or formed plates connecting the inner and outer channels—usually two at each truck; the front and rear bumper plates, the draft rigging at the back end, the draw castings and buffers at the front end, and the center pin castings where the frame bears on the trucks. A wooden platform is commonly placed on top of the frame.

XXVI. THE TRUCK. There are two standard types of car trucks—the pedestal type and the bolster type. In the pedestal type, the journal box slides up and down between jaws (the engine truck of a locomotive is always of this type) and the springs rest directly on top of the journal boxes. In this type the side frames have vertical motion relative to the track, but none relative to the car body; the truck frame consists of the two side plates, and the necessary cross bracing. The weight is carried to the wheels through the bolster, a cross frame, which may be riveted to the side frames, but is more often suspended from a rigid cross frame by swing links which gives the truck a considerable amount of side play, which may be very desirable. In the bolster type truck, the journal boxes and side frames are a rigid structure; the relative motion is between the car-body and the side frame. Weight is carried to the bolster as before which extends out through the side frames, and under the bolster the springs are placed, these resting on the "spring plank" which is usually a channel set open side up and riveted to both side frames. A small amount of side play can be obtained with this type of truck by the tipping of the springs, and sometimes rollers are provided between the bolster and the springs.

XXVII. THE BOLSTER. The bolster is made of a single casting, or a combination of structural members. Normally it carries one half of the tender weight at the center, but in rounding a curve or with an unequelly distributed load the side bearings may come into contact and a portion as much as one half of the weight on the truck in extreme cases may be carried on the side bearing. This requires that the bolster be strong enough for either loading condition; there is the additional condition affecting its strength on the opposite axis that it must be rigid under the most severe retarding conditions to be encountered

XXVIII. THE SPRINGS. Both coil and leaf springs are used in car trucks, and frequently the same truck will employ both varieties. Leaf springs are commonly used in pairs with the concave sides turned together—called in this case "full elliptic" springs. There is only one fundamental difference in the design requirements: the spring must serve its purpose the when car is heaviest without going solid and when the car is lightest without allowing the box or the bolster as the case may be to stand against the stop, the spring being too stiff to deflect under the light load.

XXIX. THE WHEELS, AXLES, AND JOURNAL BOXES. The standard car wheel in China is 42″ diameter, but recently on freight equipment and on tenders there has been a tendency to use smaller sizes, usually the 36″ wheel. Several types of centers are used : the solid type, rolled or cast; the spoke type, where the hub and rim are cast-welded around wrought steel or iron spokes. In general separate steel tires are used, held in place by shrinkage and retaining rings. Some of the newer cars with smaller wheels have tires cast integral: when the final turning has been made the original tire may be turned off entirely and a standard steel tire applied. The standard journal sizes are 4 × 8″, 5 × 9″ and 5 1/2 × 10″, the latter two corresponding to American standards, while the first is a standard British axle. The journal box used is substantially the same as the American M. C. B. box, provided with a screw cover in place of a hinged lid. The journal centers are 75, 76 and 77″ respectively for the three sizes. Either wool waste packing or oil-pad lubrication may be used in this type of box.

XXX. THE BRAKE RIGGING. Truck brake rigging is generally arranged as shown in Fig. 17, whether hand or power brakes are used. The majority of goods wagons in China have only hand-brakes, and the brakes at the two ends of a car are frequently independent. Passenger cars (at least all of those having bogies = 4-wheel trucks) are provided with power brakes as well as hand brakes. The necessity of arranging the two brakes so that one will not interfere with the other is an interesting problem, usually solved by connecting the hand-brake to the leverage though a chain, which of course can transmit no compression. In order to secure long enough live and dead levers (the two levers B and D in Fig. 17) they are usually slanted at an angle of about 45 degrees relative to the bolster. It is also important to note that in order to have the pull on the two beams exactly the same it is necessary to make the leverage on B slightly greater. (Why?) All of the rods should have length adjustment in order to take up the wear in the brake shoes without increasing the piston travel.

## LOCOMOTIVE DESIGN

XXXI. MOTIVE POWER SELECTION. Locomotive design is one phase of the larger subject of selection. The problem involved is this: the *most suitable* motive power is to be provided for the service under consideration: in a more specific way we may say that the engine which will give the best *commercial efficiency* is to be provided. In a given case this might mean the provision of a second-hand locomotive purchased from another road; it might or it might not mean the provision of a machine with the most advanced fuel saving equipment ; it might involve the provision of ten engines where five were contemplated or vice versa. To get the most and the best locomotive service at the lowest cost, both present and ultimate costs being considered: this is the problem. This brings the question into the realm of economics, and involves the great questions of capital, interest, depreciation, running expenses, labor, supplies, fuel and water, effect on the track, and many others.

XXXII. FIRST COST. During that past few years the price of a steam locomotive has ranged from fifteen to thirty cents gold per pound of locomotive alone, ready for service, the price including the tender also. Multiply this figure by the exhange ratio and add about 20 percent to cover ocean freight, insurance and reerection in China and an idea of the cost in this country is obtained. The highest prices in America were accompanied by the highest value of the silver dollar, so the highest prices yet paid in terms of silver will not be greater than about 50¢ per pound. Men who have had ample opportunity to observe state that the engines which have been built in this country have cost much more than this figure: construction in shops designed for repairs is always very expensive. Capital for railway improvements in China comes from two sources: in the case of profitable lines from income; in the case of unprofitable lines, from loans or government advances. Locomotives bought out of income present a peculiar problem in business economics, for with the conditions prevailing in China, no interest needs to be charged against them. Where capital comes from the Ministry in the form of government advances, the interest is written into the accounts at five percent. Money which has be actually borrowed outside for such purposes has cost from 8.5% net up.

XXXIII. DEPRECIATION. Every well conducted business sets aside in one form or another a depreciation reserve to cover the wearing out of equipment, and a railway should be no exception in this matter. Depreciation is of several varieties:

Absolute depreciation,—this is purely deterioration. It is closely connected with the question of repairs; theoretically could be reduced to zero, and in fact, if repairs include improvements in the case of a locomotive, the absolute depreciation may become negative.

Relative depreciation—this is a measure of the adequacy of the machine for the service. If traffic conditions remained practically constant, year after year, relative depreciation would be almost identical with deterioration, but if train loads and speeds increase and more economical engines can be built to do the same work, relative depreciation may become the paramount factor in the account.

Artificial depreciation—this is really a question of profit and loss on first cost, but it is convenient to consider it sometimes as a form of depreciation. Engines brought in to China in 1919 are now worth in terms of silver practically twice as much as they were when new—here is a case of *negative* artificial deprsciation.

Arbitrary depreciation—this is the book-keeping figure in which account is taken of all of the others—a flat percentage charged off of the value each successive year, a "depreciation reserve" charged againt the company's surplus.

XXXIV. OPERATING COSTS. These consist of fuel, water, supplies, and labor. Coal mainly of indifferent quality, for railway use ranges in price from $3.50 to 32.00 per ton delivered on the line which is to use it—hence to this cost must be added a freight charge. The average figure for the C.G.R. is about $7 per ton, highest prices being on the Shanghai lines (around $20) and on the southern lines. Here coal consumption is the all important part of the cost, and a difference of a hundred pounds a day may govern the entire question of selection and design. There are no available figures for water costs—fifty cents for a tank-full is a common though purely arbitrary inter-line charge. Supplies—that is, oil, waste, and tools, form a considerable item. Labor costs include the wages of two (sometimes three) men, the engineman drawing as high as $2 per day, the stoker sixty cents to a dollar. Wages are constant regardless of the type or weight of the engine.

XXXV. REPAIRS. In China, all work which is done on a locomotive is classed as repairs, even if this involves a complete rebuilding and altering of the wheel arrangement. Various railways report figures of from 6 to 20 ¢ per locomotive mile for repairs. Obviously the larger the engine the larger the repair bill, though the increase is not by any means in direct proportion. Tender repairs are included—approximately two cents per mile. These figures include "over-head" charges, supervision, cost of maintaining shop buildings and machinery, inspection, and all of the costs of the locomotive administration.

XXXVI. OTHER CHARGES. There are many other charges which have some effect on the locomotive economic problem. These may be enumerated without further comment: maintenance of track and structures; terminal construction costs; cost of coal and water service; freight rates; car loadings, costs, and depreciation charges.

**XXXVII. LOCOMOTIVE DESIGN.** Let it be assumed that traffic conditions demand additional motive power and it is proposed to purchase new engines. Several prospects are open:

1. The duplication, absolute or partial, of existing power.
2. The construction of new power in such a manner as to give a maximum number of parts duplicating present equipment.
3. Construction without reference to existing equipment.
    a. Complete econmic specfications, and liberty to the designer in the selection of all details.
    b. Partial specifications, such as type, cylinders, etc.

The conditions are obviously more or less overlapping. No. 1 leaves nothing to the designer, unless some minor device on the previous engine has been a failure and he must replace it with a better. No. 2 can be carried out to a considerable degree between certain types of engines—for example a ten-wheeler and consolidation, or a Pacific and Mikado respectively may have all of the following parts duplicated:

Boilers and all boiler fittings complete except angle irons on the under part of the boiler for braces to frame.

Trailing trucks, complete, except radius bar.

Axles, journal boxes, horn—blocks (if plate frame is used) and pedestal fittings.

Cylinder castings (if stroke is not more than 2″ different)

Cab and fittings.

Crank pins, crank pin bushings, valve gear details.

It is much better if this is to be done to design the two engines at the same time, than to have to fit details used on one to the other, especially if the second engine was not contemplated when the first was built.

Case 3a is a rather unusual condition, but a very desirable one. The necessary information which must be furnished involves all of the operating requirements, the cost of coal, terminal conditions, (especially stall-lengths and turntable-lengths), and much other incidental information, the possession of which is an index of the usefulness of the designing engineer to his employers, ether the railway itself or the locomotive builder. In this case the general process is as folows :

1. From the operating requirements develope a skeleton speed-pull curve, and fit to this suitable cylinders, drivers, and working pressue, and a minimum boiler capacity, as represented by the smallest weight which will give the required coefficient of adhesion and the smallest boiler which will deliver the required amount of steam within the firing capacity of the average stoker.

2. It will be found in general that the size bolier thus determined does not tax the driver weight allowance and that a larger boiler may be used, over the required number of driving wheels. This may be still further enlarged as the result of using additional drivers or trucks, though the use of additional drivers does not commend itself.

3. It now must be learned by means of an economic investigation of cost, interest, and operating expense whether the first boiler determined or a larger one is the most economical : that is, whether improved thermal performance on the road will compensate for additional first cost and additional repairs.

4. The final test of each modification of the design is a weight estimate and distribution. After one has been made, others can be made by comparison and much labor saved.

The last case named-(3b)—is much commoner, as many details of the new design will be settled by the railway before the completion goes to the designers, hands. Probably the commonest specification of all is this: "Wanted a 462 type, with 20 × 28 cylinders, 69″ divers, 180 lb. boiler pressure, and a driver weight limit of 37,000 lb. per axle." Given information of this sort it is the province of the designer to produce the best machine possible within the specifications, and what he can not control no one can hold him reponsible for.

XXXVIII. THE WEIGHT ESTIMATE. The weight estimate is made up on the basis of accounting for about seven-eighths of the total weight, the parts thus accounted for bearing a very definite relation to the total. The parts not put into the account are those such as the running boards, brake rigging, driver spring rigging, foundation brake rigging, etc. which are so scattered over the engine as to have little effect upon the center of gravity. The process of making the distribution has been already explained. The parts taken into account in the weight and center of gravity calculation are as follows :

The Dead Weights : The counterbalance having been already made much labor in this part of the work is already done. All parts are figured by pure mensuration methods. The parts to be taken into account for each driving wheel are : tire, wheel-center, axle, journal box, any equalizers which rest upon the boxes, crank pins, rods. For the engine truck and trailing trucks, tires, centers, axles and boxes : usually the labor of calculating the frames is, saved since total truck weights are commonly available.

The Spring Weights.

Boiler, fugure part by part by mensuration (.296 per cu. inch to include rolling allowance)-

Water, same method. (Hot water, 55 lb. per cubic foot.)

Tubes, flues, and superheater units : weights per foot can be taken from tables.

Superheater headers : 36 lb. per unit.

Throttles : 50 lb. per inch of diameter.

Dry pipes, front end steam pipes, exhaust nozzles, mensuration.

Headlight cases : 100 to 250 lb. complete.

Headlight generators : 220 to 300 lb.

Sand—boxes : mensuration. (Sand 100 lb. per cubic foot)

Fire : 30 lb. per square foot for wide fireboxes, 50 lb. for narrow.

Grate : shaking, including supports and shaking apparatus, 70 lb. per square foot.

Ash pan : 20 lb. per square foot of grate area, plus 300 lb. per hopper.

Fire—brick : mensuration, 100 lb. per cubic foot.

Cab fittings : injectors, steam and overflow piping, 140 lb. each. Steam turrets, 100—250 lb. Steam gauges, 20—30 lb. with brackets. Air brake valves, 75 lb. with piping. Fire—doors, hand operated, 60 to 120 lb. Power operated, 150—300 lb. Reverse—screws, complete, 200 to 300 lb. Throttle levers and rigging, 40 to 150 lb. Cab tools, 150 lb. Distributing Valve for air-brake, 180 lb. Make about 25% additional allowance on sum of cab fitting to cover minor details.

Safety valve turrets, 100-120 lb., valves, 30 lb, whistle 20-35 lb.

Air pumps, complete with brackets, 800 to 1200 lb. for commonly used sizes.

Frame plates or castings : (including pedestal fittings, all bracing, bumper plates or castings, and cross-ties), mensuration, based on rough sketch layouts of the parts if not yet detailed on drawings.

Cabs weigh from 1400 to 3000 lb. according to size and construction.

Cylimders complete = 25 Dl lbs.

Couplers : M. C. B. Coupler heads weigh from 300 to 500 lb. ; add shank, pocket, or carrier plates, and draft gear if used.

Brake cylinders ; 400 to 700 lb. for pair complete, bracket or support extra.

Brake cranks or levers, fulcrum, and shaft, mensuration.

Brake reservoirs, mensuration—quarter inch plates.

Engine and trailling-truck equalizers (those above and between driving wheels not included), mensuration.

Pilots, from 200 to 600 lb. according to construction.

If the list of parts given above is carried through the total weight accounted for in the dead-weight and spring-weight classes will be from 84 to 88% of the total weight of the engine. If no engine of similar design and known weight is available for a part-by-part weight comparison, this percentage method can be used with good results. In general, the low percentage goes with small scale and incomplete drawings, and the largest percentage wtih a drawing complete in all details.

It is of interest to note the ratios of each part of the weight to the total. Dead weights closely approximate 23%, regardless of type. The boiler, water, tubes, and flues combined compose about 35 to 40%.

On a Pacific type engine weighing about 170,000 lb. the "unaccounted for weight" will be about 25,000 lb. The main elements in this will be the following: springs and equalizers, 3000 lb. Running board 1000, hand rails, steps, etc., 500 lb. Lagging and jacket for boiler, 2500 lb., air brake piping 2000, foundation brake rigging 3000, etc.

# Notes on Railway Mechanical Engineering

## PART III.

## ADMINISTRATION OF THE LOCOMOTIVE DEPARTMENT.

I. ITS PLACE IN THE GENERAL ORGANIZATION. There is much variation in the organization of railways in this matter abroad, but in China the arrangement is apparently invariable. In place of being the equal of the Chief Engineer as on most lines abroad, the Locomotive Superintendent reports to him. There is a definite reason for this, growing out of the method by which most of the goverment railways were originally financed. The Chief Engineer naturally took charge of all operation during the construction period, and as the time of operation came, he was not willing to relinquish any functions he could retain.

II. THE GENERAL OFFICE. At the head of the Locomotive Department is the Loco Superintendent, responsible to the Chief Engineer, and he in turn to the Managing Director. The Loco Superintendent on the larger roads is provided with one or more assistants: these men may be designated to certain portions of the work where they have actual supervision or they may be merely general assistants, taking such work as from time to time they may be directed to. A typical arrangement of assistants with specified duties might be:

One with direct supervision over the actual operating, superintending the distribution of locomotives, the employing and training of crews, the assigning of runs, etc.

One with supervision over the terminals. ("Chief Inspector")

One wlth supervision over the central shops. ("Works Manager")

One with supervision over the strictly engineering work of design, remodelling, performace etc. (Mechanical Engineer or Chief Draftsman)

One with general supervision over personnel.

In addition there are two very important officials connected with the Loco Department who do not report to the Loco Superintendent: the Storekeeper, responsible to the Chief Stores Manager, and the Auditor, responsible to the Chief Accountant or General Auditor.

To the first of the assistants listed, or rather to the first two combined, for all of the goverment railways place these two functions in the hands of one man, are responsible the several "Local Inspectors" at the various engine terminals. These men are not *inspectors* in any accurate sense, but are in reality general foremen, exercising control over three distinct groups of men: first the terminal crews, responsible for the "turning" of the engine—the actual turning, also inspection, water, oil and coal supply, cleaning, dumping and making fires, washing boilers, etc.; second, the terminal mechanics who make such running repairs as can be made in the engine sheds; third, the engine crews themselves.

The duties of the "Works Manager" are obvious from his title. Responsible to him are the foremen of the several shops- Erecting, Boiler, Machine, Smith, Foundry, Carpenter, Paint, Tank, Truck, etc., and the necessary labor gangs which are part of every shop.

The Chief Draftsman is responsible for the conduct of the drawing room and related work: he becomes a very important official on a line which does construction or rebuilding for itself. He is generally charged with other work which can be handled by a man with high class technical training: material tests, specialty and general performance tests.

III. TYPICAL SHOP ORGANIZATION. In a large shop the lower organization is approximately as follows:

Assistant Manager, in Charge of Loco. Departments:
  Erecting Shop Foreman, gang foremen, mechanics, helpers, laborers.
  Machine Shop Foremen, division foremen (wheels, rods, valve gear, frames, etc.), mechanics, helpers, laborers.
  Boiler Shop Foreman, division foremen (riveters, flangers, layout men), mechanics, helpers; laborers.
  Tender Shop Foreman, division foremen (tank, frame, trucks).

General Labor Foreman, sub-foremen, labor gangs.

Office force, including time keepers, clerks, etc.

Assistant Manager in Charge of Car Department·

Foremen in charge of truck shop, steel shop, paint shop, brake shop, carpenter shop, cabinet shop, upholstery shop, etc., with same subordinates.

Labor Gang Foremen, sub-foremen and laborers.

Stores Manager (serving both the shop and the terminal, but responsible to the General Stores Manager.) Storekeepers, laborers.

Accounting Department, same relation as stores department. Auditors, accountants, clerks.

IV. TYPICAL TERMINAL ORGANIZATION. With the Local Inspector at its head, who will in general take direct oversight of the conduct of the engine shed, the subordinate organization is as follows:

The various divisions of the engine shed work will be in charge of working foremen, commonly styled in China "Number Ones". There will be such a man in charge of each group: mecheanics (fitters), cleaners, boiler-washers, laborers.

Outside work is carried on in a similar way, in charge of an Assistant Inspector, with Number Ones at the turn-table. coal dock, oil-house, ash-pit, and over floating labor-gang.

Office: such clerks, recorders and time keepers as needed.

The general storesdepartment and accounting department serve the terminal organization as well, but are in no way responsible to it.

## LOCOMOTIVE OPERATION AND MAINTENANCE

V. LOCOMOTIVE PERFORMANCE. The term "Locomotive Performauce" while sometimes used to denote merely the thermal performance while in actual service, hauling a train, will here be discussed with regard to its more general meaning—that is, the total service which the locomotive renders. A locomotive is an expensive machine to buy, to maintain, and to operate, but it is much more expensive when once it is in the possession of the railway if it is not operated economically. However, it is fair to remember that economical operation of the entire locomotive stock of a line may mean laying part of those engines up in storage (called commonly "in white-lead"), but with the small number of locomotives in China it does not seem probable that this condition could occur. The total service of a locomotive is measured in terms of the number of ton-miles which it hauls in a year, in the case of a freight engine, and in this total there are several factors. The first is the *number of tons per trip*. "Tonnage rating" as practiced in the United States is unknown in China for several reasons; as a rule the locomotive on the C.G.R. are not loaded to the limit of their capacity; the lines are level, and even on those lines with considerable coal traffic, a train which can barely be dragged is not economical. Another factor which tends to limit train lengths is the lack of power-brakes on the cars, and no amount of inspection or instruction can make a hand-brake system effective, mainly becasue of the indifference of the trainmen. It is not known that any investigating has ever been done with the idea of determining economical train-loads and corresponding speeds, except possibly on the Peking-Suiyan where the matter of getting cars over Nankow requires close study. The question of *speed* in itself has little to do with the total transporation output of the locomotive per year, because no ordinary variation of speed would permit any additional trips, especially since the runs are usually assigned. (See later discussion.) The length of the division or district is of great importance however: the Chinese railways have had to serve the important commercial centers as they found them, with no control over their location, and this has resulted in a very haphazard set of division or district-lengths and goods-locomotive runs—from 75 to 200 miles. For any set of operating conditions there is doubtless a most economical length of run, and an increase beyond this figure will not increase the annual performance—on the other hand may actually decrease it. Unfortunately this is a factor which can not often be worked out in time to be of service, but common sense limitations in freight service put the distance below 200 miles, unless the track be very level and the loading very light. The third important feature is the number of trips the engine can make in a year. There are two principal factors controlling this quantity : the assignment of runs, which limits to one trip or one round-trip a day, and the terminal service, which controls the number of days of service lost in the terminal and the shop. If the terminal is well equipped so that minor repairs can be made promptly, the number of days lost is reduced to a minimum. The following figures show the performance obtained from the goods engines on the six most important C. G. lines (1920 figures).

| Line | Total Ton Kilos. | Goods Locos. | Ton-kilos per year per loco, |
|---|---|---|---|
| P.H. | 1,379,000,000 | 90 | 15,300,000 |
| P.M. | 1,048,000,000 | 127 | 8,300,000 |
| T.P. | 1,185,000,000 | 68 | 17,500,000 |
| S.N. | 304,193,000 | 15 | 19,000,000 |
| S.H.N. | 72,569,000 | 12 | 6,050,000 |
| P.S. | 210,184,000 | 31 | 6,800,000 |

If the figures above are further modified they will have more meaning : the average goods train on the C.G.R. is about 300 net tons, assuming the average freight run as 166 kilos, a single train hauled represents 5,000 ton-kilos—hence if five ciphers are dropped from the "ton-kilos per loco. per year" and the result doubled we have a *very rough* measure of the number of trains hauled per year.

VI. ASSIGNED AND POOLED RUNS. When a certain crew takes a certain engine out on a certain schedule every day or at other fixed intervals, the run is said to be assigned. When a train is "called" and the suitable engine which has been in the house the longest is called for it, and the crew which has had the longest rest is called, regardless of attempts to keep engine and crew together, the runs are said to be "pooled". The assigned run is universal in China. Some of the advantages are :

Running regularly on a given engine the crew will take more interest in their engine, making minor repairs and adjustments themselves, and interesting themselves generally in its maintenance in a way that is not usual on pooled engines.

Running regularly on the same schedule, familiarity with every detail of time, meeting places etc., is a factor of safety.

Every locomotive has certain idiosyncrasies and peculiarities, and it is an undoubted advantage to have a crew familiar with these.

Advantages of the pooling system may be named as follows:

In general a locomotive requires much less terminal time than the rest time of the crew, hence fewer engines are required to perform the same amount of transporation.

The crew does not lose time because its "own" engine is held out for repairs.

The number of reserve engines is less than when runs are assigned.

VII. THE LOCOMOTIVE TERMINAL. The locomotive terminal is charged with the functions commonly called "turning" the engine. The various parts of this process are as follows: receiving the engine from the traffic department, usually at the receiving track of the terminal; providing men to move the engine and care for the fire and water as long as the engine is under steam; cleaning or dumping the fire; disposal of the ashes; provision of fuel and placing it on tender; inspection for defects, loose parts, etc.; minor repairs to make good such defects; cleaaning; boiler washing; housing; turning; finally filling the; tender-and delivering to the transportation department ready for service. The functions will be examined separately.

VIII. THE ENGINE HOUSE. Two main varieties of engine house are in use, those provided with radial tracks, radiating from a turn-table, and those in which the tracks are parallel. Both are in use in China, the radial type representing German, American and Japanese practice, the rectangular type representing French and Bristish practice. There are clear advantages for each type of house, some of which are as follows:

Advantages of the Radial-track type :

Economy of total space, provided auxiliaries are grouped closely around engine house.

Accessibilty of every engine in the house.

Better lighting and ventilation, espcially at the head of the engine where most repair work must be done.

Adaptability to drop-pit arrangements.

Ease of oversight and supervision.

Cheaper and more efficient smoke removal.

Convenient location of the turntable.

Advantages of the parallel-track house :

Less space per engine for actual storage.

Better adaptability to crane installations.

Less first cost for given number of engines.

Easier to get engines out in case of fire.

Many other advantages can be given, most of which definitely count in favor of one type and against the other type.

The engine house should be built of non-inflammable and non-corrosive material. Concrete is the most satisfactory : creosoted wood is also very good for the interior construction, especially overhead trusses, inside of roof, etc. The roof should be covered with some substance resistant alike to sparks and gas-fumes : various forms of prepaired roofing with gravel covering are doubtless the best. The floor should not be of cement, as it breaks badly under dropping of tools and similar abuse. Wood-blocks, tarred and set with the grain vertical make a very durable floor, especilly if set on a concrete base. Ordinary rammed earth makes a good floor, though objectionable if wet. In any case a heavy plank should be provided just outside the rails and full length as a base for jacking. The pits between the tracks should be continuous ; the bottom of pits should be crowned, so as to drain to one side ; all pits should be provided with water-pipes, compressed air, and outlets for electric torches and drills. The best method of heating and ventilating combined is the forcing of an air-blast up through the pits, and out the smoke jacks ; this blast may be heated in winter and cooled in summer. Steam heat is less satisfactory.

and it is very undesirable to place steam heat pipes in the pits, on account of the danger of burning the men at work there.

Modern engine houses are provided with a small lay-out of machine tools, by means of which much light repair work can be done, thus saving the time otherwise required to get the engine into the shops. The minimum equiment may be considered to be two lathes, one large enough to take the piston rod, and one of small size for pins, etc. ; a pipe threading machine, drill press shaper, two forges, and necessary bench equipment to correspond. General tool equipment for the house cosists of air-motors, electric drills, wrenches, reamers, taps. etc., hand trucks, sledges, (including a number of soft faced hammers), and a generous supply of jacks, ordinary, traversing, and hydraulic.

For a roundhouse the turntable may be either hand our power operated—this is purely an economic question. For a rectangular house, the track approaches should not involve any curves sharper than 22 degrees, and if possible, keep within 20 degrees curvature.

IX. COALING ARRANGEMENTS. The only coaling arrangments in use in China are minor variants of the familiar basket and elevated platform method, and with labor at current prices there is no economic reason for any elaborate power-operated coaling device, except under special conditions noted later. Much improvement could doubtless be made in the present method by giving thought to the use of elevated tracks, hopper-bottom cars and the like. Where labor is relatively much more expensive, coaling stations are not uncommon where one man with mechanical aid delivers coal to several engines per hour. The general principles of such a station can be easily seen in advertising matter in the technical press.

X. ASH PITS. The only method of ash removal practiced in China is the coolie and his basket—and from four to eight handlings of the ashes before their eventual disposal. Here again there is little economic warrant for the use of power-operated devices, but the disposal as at present effected certainly leaves room for minor means of labor saving that are worth careful consideration. Deeper ashpits, with sloping bottoms which would permit the cinders to slide directly into hand—operated elevators is one suggestion which has merit.

X1. OIL STORAGE. On account of its inflammability oil should be stored in underground tanks tapped by measuring pumps. This department can be conveniently combined with a room for the storage of the enginemen's personal tool—kits, lanterns and clothing.

XII. THE SIZE AND CAPACITY of a TERMINAL. The determination of the size of a new terminal as indicated by the number of engine stalls in the house, for example, is a problem difficult of solution because of the many unknown quantities. A standard method is the carrying through of the actual schedule with dummy "trains" and allowing the usual proportion of delays and minor accidents—this plan has given good results in the design of notable terminals. However, most terminals are not designed but simply evolve, and when an old structure is replaced by a new one, there is a very definite knowledge as to what the capacity require ments are. The track layout is a feature which affects the capacity almost as much as the mere space allowance. The ideal terminal is one in which the locomotive could travel continuously in one direc tion, and the one which is actually most efficient is the one in which the fewest reversals in "direction of flow" become necessary. The track layout should be so designed that there is no place int he terminal where the blocking of a track by accident would interrupt the action of the plant. This is of course impossible to accomplish in the "inner circle" of a roundhouse, where an "engine in the pit" is the foreman's nightmare.

XIII. THE RAILWAY SHOP. The principal divisions of the railway repairshop have been noted under the heading of "Administration." The locomotive group in general consists of four principal shops—the erecting floor, the machine shop, the boiler shop, and the smith shop. The car division includes the tender shop, truck shops, steel car shop, smith shop, carpenter shop, and minor shops for cabinet work, upholstery, painting, brake work, etc. There will be a foundry serving both divisions, also a general store serving both divisions. There is a tendency in this country to group several of the shops under the same roof—this a questionable economy. Among the general features of the shop layout which should be mentioned are the following:

Transportation of heavy parts; ready and convenient means must be provided for moving boilers to and from the the erecting floor and boiler shop, and wheels and trucks from under the engines to the machine shops. The entire system of shops should be interconnected by standard gauge tracks with curves such that a four-wheel switcher can push cars of material in and out of any of the buildings, and to the storage yard. The service tracks should run under the cranes in every building to permit convenient unloading and loading.

Heating, lighting, sanitation: a cold shop in winter is a poor form of economy and an adequate heating system will pay for itself in more and better work from the men. Good light is an absolute necessity for good work—the greatest possible use of daylight should be used, and after that a carefully studied system of electric lighting. Little can be said of sanitation, except tbat a general principle may be laid down: sanitary conveniences must be made sanitary and kept sanitary, usually in spite of utter indifference on the part of the workmen to their own health and comfort; at the same time such conveniences must not be made comfortable enough to provide loafing places. The use of some form of disinfectant with an oder that becomes obnoxious in a short time is one solution of this difficult problem.

XIV. THE ERECTING FLOOR. The central feature of the shop layout is the erecting shop. Two distinct types of floor layout are used: the longitudinal system and the transverse system. Under the former arrangement, the erecting shop has three parallel tracks, of a length which determines the capacity of the shop: the central track is used for service only. In this case an engine is lifted by two cranes. In the transverse shop there are a number of parallel tracks each long enough for a single engine, and an engine is lifted by one crane only. In either case careful thought must be given to methods of unwheeling and rewheeling the engine, and of getting the boiler off and into the boiler shop. As to crane capacity, while it is desirable to have the cranes (whether the engine is to be lifted by one on two) capable of lifting the largest engine complete with wheels, the largest normal requirement is the lifting of the engine without the wheels, and also of course without water, fire, sand, etc. Practically two thirds of the weight of the heaviest locomotive should be the crane capacity. However, the locomotive can also be erected by first setting the frame on the wheels, then the boiler on the frame, though this requires its occupation of the service track much longer, Under the latter condition a total crane capacity of one third the weight of the engine is required. The use of high—lift hand—power jacks is favored in this country as a method of unwheeling. To be satisfactory the jacks are of very large size, and either a complete set of four or six is required at each pit, or else the difficulty of moving them is involved. In either case since the crane is required anyway it is a duplication of equipment. After the questions of unwheeling and the transportation of the heavy parts are considered most of the arguments in favor of each type of shop become obvious. It would seem that for shops with less than about ten pits, the transverse or cross—shop with transfer table would be the most satisfactory, but for larger shops the longitudinal building has many advantages. The number of engines to be provided for (number of pits) requires careful consideration. From seven to ten percent. of the time of a locomotive may be spent in the shop——hence the number of pits should be from seven percent. to a somewhat higher figure than ten precent., (to account for accidents) of the total number of engines served. A small pit-ratio means a long line of engines held outside, waiting for repairs. The floor-space in the erecting shop proper should be about 1750 square feet per pit, including aisles, service tracks, etc. The pits should be continuous, and provided with drainage, air, and light outlets. The shop should have an ample supply of hand-trucks, jacks, bars, sledges, soft-face mauls, etc., storage racks of ample capacity to keep the floor clear of miscellaneous material, air motors, drills, riveters, hammers, and electric drills.

XIV. THE MACHINE SHOP. The machine shop should be fully equipped with telferage fittings, and should have a service track with direct connection to the erecting floor. The floor space should be about fifty percent greater than that of the erecting floor, say 2500 square feet. The tool equipment of the shop is a matter which varies considerably with the designer, and it is difficult to extract anything very generally applicable from the inconsistent tool-layouts there are to be found in most shops. The following approach seems the best: if there are from eight to twelve modern tools per pit, there will be no waiting on the machine shop for output. Take a one-pit shop as an example: say ten tools are allowed, and the selection would be as follows: two lathes, 1 planer, 1 milling machine, 1 shaper, 1 grinder, 1 drill, 1 press, 1 hacksaw, 1 bolt cutter. Now the last three in this list would not need to be duplicated if there were two pits, but the other seven machines are required in direct proportion to the number of pits. This allows three machines of a miscellaneous nature per pit; it is natural that when four lathes are allowed they will be of assorted sizes, and so with the other machines. On this basis the tooling for a six-pit shop would be as follows:

12 lathes: 1 driver-lathe 84″; 1 car wheel lathe, 48″; one automatic screw machine, 1 centering machine, 1 stay-bolt lathe, 1 precision-lathe for tool room; 2—30″ swing × 10′ bed lathes, 1—8′ × 24″, 2—6′ × 20″, 1 turret lathe.

6 planers: 1—72″ (between housings) × 72″ (maximum height tool from bed) × 18′ (length of platen), 1—48 × 48″ × 15′, 1—36 × 36 × 10′, 1—24 × 30 × 10′, 1—18 × 18 × 6′, 1 link planer.

6 shapers: 1—30″, 1—24″, 1—18″ crank shapers, 1 double traversing shaper, 1 worm-drive (long stroke) shaper.

6 milling machines: (somewhat excessive) 1 No. 3, 1 No2 1/2. 2 No. 2, Universal; 1 No. 2, 1 No. 1 plain Milling Machines.

Grinders: 2 cylindrical grinding machines, for piston rods, etc., 1 small grinder for twist drills and other small work, 1 grindstone, 3 emery wheels.

6 Drills: 1—6′ radial drill, 1—4′ radial drill, 2—24″ drill-presses, 1 bench drill, 1 sensitive drill in tool room.

Miscellaneous machines, (total 18): 1—60″ boring mill 1—48″ boring mill, 1—24″ boring mill, 1—24″ slotter, 1—18″ slotter, 2 boring-bar equipments complete, 2 hack-saws, 1 bushing press, 1 axle and crank pin press, 2 double head-bolt cutters, 1 slab-miller. 1 portable crank-pin-turning machine, 1 portable welding outfit, 1 portable valve seat grinder, 1 pipe-threading machine.

XV. THE BOILER SHOP. The space alloted to the boiler shop should be only slightly less than that of the erecting shop. Machine tool equipment therein consists of portable riveters, gang drills, flange-machines, edge planers, and bending rolls. Wide storage space under cover and convenient means of handling heavy parts, including completed boilers are necessary.

XVI. THE SMITH SHOP. The Floor requirement is about three-fourths as great as the erecting shop. The tool equiepment consists of power-hammers, forging-presses, drop-forging machines (die presses), and an equipment of forges of various sizes. Power ventilation including suction hoods over the forges is considered essential. The tire-heater is frequently placed in the forge shop, and this is unobjectionable provided a convenient method of handling the wheels is available. The punches and shearing machines are usually placed bere.

XVII. THE CAR SHOPS. The machine shop of the car division has many points in common with that of the locomotive division, but the work should never be mixed, on account of the generally lower standards which prevail in car work. Special axle lathes are provided, and boring mills of suitable size to care for wheels and tires. One or more wheel-lathes, accomodating the axle and two wheels comingin for tire-turning are necessary. In the steel car body shop should be punching, shearing and edge planing machines, also portable welding sets, drill presses, and a device consisting of parallel beams pressed together by jacks which can be used to straighten an entire car body damaged by accident. The equipment of the minor shops can be well passed by as obvious.

XVIII. LABOR SAVING TOOLS AND MACHINERY. Some work on railway equipment can only be done by machinery—much other work can be done by hand or by power. There are many peculiarities of the question, which is one of combined humanitarianism and economics. It is well to remember however, in considering the first phase that the doing of work by inefficient methods merely for the purpose of giving employment does not ultimately do any good, though it may appear to do so if only the case in hand is considered and the view is a narrow one. There are three general arguments in favor of production by modern labor-saving methods : cheapness, quickness, and quantity. The first of these may or may not be effective in any special case, and only careful analysis can determine the answer. The latter two rarely are met with at present in studying conditions on the Chinese railways : the production of transportation is carried on at so slow a rate, and with a plant comparatively ample that no means of increasing the production is in urgent demand. These generalities may be thus illustrated : a terminal the size of that at Shanghai North Station handles about forty engines per day ; in the same size terminal with every modern improvement, four times this number could be handled. But as long as *forty* and not 160 is the demand, and the devices will not save enough in labor costs (due to low wages) to cover interest and depreciation upon them, why install them? the time may come when in order to cope with a heavier demand, additional land must be bought—*or* the terminal be partly "modernized"—in that case, a hundred thousand dollars worth of improvements may save a million dollar investment in land—and the time of the modern devices will have arrived.

XIX. COSTS OF RAILWAY OPERATION. The following figures are selected from the data in the 1920 report of the government railways to give the student an idea of the costs of various items. The figures are selected in such away as to make them representative as possible, omitting results from those lines which give returns which for very well known reasons are meaningless.

Operating Expense per Train Kilometer $1.50, made up as follows:

General 26¢ (general administration expenses)

Traffic 17¢ (station, unloading and loading, shunting, etc.)

Running 34¢ (actual running of trains)

Maintenance of Equipment 37¢ (Shop and terminal expenses)

Maintenance of Way 36¢ (maintenance of track and structures)

Average load per train, 300 passengers, 300 tons goods.

Analysis of Running Expenses : (per train kilometer)

| Total Running Expense | Cost of Coal per T.K. | Coal Cost % | Railway |
|---|---|---|---|
| 22¢ | 10¢ | 45% | Cheng-tai |
| 31¢ | 15.6¢ | 50% | P.M. |
| 47¢ | 36.2¢ | 77% | Shanghai Nanking |

Cost of lubricants for engine and train each 1¢ per train kilometer.

Maintenance of Equipment :

Per train kilometer, 32¢ ; per loco kilometer, 6.4¢ ; per wagon per year $120 ; per carriage per year, $500.

Kilometers run per year for road locomotives : S.N., 66,880 : P.H., 57,000 : S.H.N., 55,500 ; T.P., 45,230, P.M., 41,800, P.S. 32,470.

Average kilometers run per carriage, 45,000. Per Wagon, 15,000.

Standby losses and fuel for shunting : Average cost of fuel per train kilometer 20¢ ; per locomotive kilometer (including arbitrary allowances or kilometrage for shunting and standing in steam), 13.6¢.

Maintenance of Way, Structures, and Track.

Average total $2000 per kilometer.

Average for Way and Structures, $1400.

Average for track, $600, distibuted as follows :

Labor, $210.

Rails and Fastening, $20 (no renewals)

Ballast, $18

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